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## XVI.

# AN ANALYSIS OF THE RESULTS OF SOME EXPERIMENTS MADE WITH THE ORVIS SYSTEM OF INCREASING THE ECONOMIC VAPORIZATION OF STEAM BOILERS AND OF PREVENTING SMOKE IN THEIR FURNACES.

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The Orvis system of preventing smoke and increasing the economic vaporization of water in steam boilers, consists in injecting by means of a large number of ordinary injectors of small diameter, numerous streams of steam, and of air induced by the steam, into the mass of hot gases of combustion filling the furnace above the incandescent coal upon the grate, these streams being inclined at different angles, horizontally and vertically, so as to penetrate the whole mass of gases in the furnace and thus to thoroughly mix them with each other and with the induced air, which latter is, of course, additional to the air received through the grate.

The injectors delivering the steam and the induced air are of iron, and the steam supplying them is taken from the boiler and

is used at the boiler pressure; whatever weight of steam, therefore, may be thus used, must be deducted from the vaporization in order to obtain the commercial result. The application of this system to any boiler does not require the least alteration in its construction or in the masonry of its setting. All that is needed is a sufficient number of injectors per furnace, and the pipes conveying the boiler steam to them.

The Orvis apparatus is one of many similar in purpose and in plan. Some have supplied steam alone in streams, and others have supplied compressed air alone in the same manner, the object of all being partly chemical and partly mechanical, namely, to furnish oxygen to the coal gases in the furnace and to mix it with them.

When steam alone was employed, the supposition was that it would be decomposed by the furnace heat and the chemical affinity of the carbon of the coal into its constituents oxygen and hydrogen, and that the re-combination of these gases with the carbon of the coal and the oxygen of the air would produce more heat that was absorbed by their decomposition by an amount equivalent to the affinity of the carbon. Something was also hoped for by the mixing of the gases in the furnace caused by the passage of the steam jets through them.

When compressed air was used alone, the object was to supply the furnace with additional oxygen and mixing.

The Orvis system is a combination of both air and steam, the latter being used to inject the former instead of a compressing pump.

The ordinary means of admitting air to the coal gases is by small holes in the furnace door, or, additionally, in the furnace front, and the air enters in virtue of the difference between its density and the density of the gases in the furnace. Sometimes the air is similarly admitted at the bridge wall.

In the earlier days of steam engineering, the most exaggerated reports were current of the great economic gain following the use of the latter plan, as shown by experiment, but later trials made by persons competent as well as disinterested, have demonstrated the falsity of these statements. Obviously, the small

quantities of air entering under such conditions, could not produce much effect under any circumstances; and even if all the oxygen of the air thus admitted were burned, still the loss due to the great mass of nitrogen accompanying the oxygen, would be serious enough to nearly neutralize the gain.

The whole case hinges on the single question answerable by only chemical analysis: Do the furnaces of steam boilers fired in the usual manner, and having the usual thickness of bed on the grate (from 4½ to 9 inches), contain any coal gases not saturated with oxygen? As far as chemical analysis has been applied to the gases of combustion in the chimneys of steam boilers, the answer is no; and, still further, these analyses (and the writer has about 500 before him,) all show that the dry chimney gases contain, as an average, 10½ per centum of free oxygen. If this proportion of uncombined oxygen exists in the chimney gases, it certainly does in the furnace gases, and shows the utter hopelessness of injecting more oxygen there with the view of making an economic gain. Practical experiments on the large scale prove the same fact, the average showing no gain or rather a small loss to result from the admission of air above the fuel on the grate, and the small gain given by a few experiments is probably due to accidental errors of observation or, more probably, to unsuspected difference in the quality of the coal consumed, in the rate of its combustion, or in the skill of its firing. The 10½ per centum of free oxygen in the gases of combustion, means that twice the oxygen chemically necessary to combustion has been admitted. This excess of oxygen is absolutely essential to complete saturation of the coal constituents, and is also the evidence that such saturation is accomplished. The dissociation of exactly one-half of the carbonic acid and steam in the furnace at the temperature of their formation, renders indispensable twice the oxygen chemically required for the complete saturation of the coal constituents. If the bed of coal on the grate be carried sufficiently thick, relatively to the size of its lumps, as in the case of gasogens, or if the admission of air be sufficiently restricted, carbonic oxide will be formed as well as carbonic acid, and the proportion of free oxygen in the dry chimney gases will be correspondingly diminished.

The general result of all the various methods which have been tried to perfect the combustion of coal in the furnaces of steam boilers, have been something worse than failures. Steam alone has been injected below the grates and above them. Air alone, compressed or of atmospheric pressure, has been forced into the furnace gases above the solid fuel. Mixed steam and air have been similarly applied. The coal has been reduced to impalpable powder and blown into the furnace mingled with air above the incandescent coal on the grate, so as to obtain a more thorough mixture of the two. The carbon of the coal itself has been converted into carbonic oxide, and its hydrogen has been made free, and then these gases have been mixed with atmospheric air in the expectation that with complete gaseous intermingling perfect combustion would be obtained with only the oxygen chemically necessary. The expectation can never be realized, for, owing to the dissociation above referred to, there can be no complete combustion in a furnace without double this quantity of oxygen. The various attempts alluded to have been made in ignorance of this fundamental truth. With twice the chemically required oxygen, combustion is perfect on the grate of the furnace of a steam boiler as habitually fired, less will not produce this effect, and more is injurious. Good firing consists in keeping the bed of coal on the grate free of holes and refuse, and *of exactly the thickness that will pass sufficient air to supply twice the oxygen chemically necessary.* When this thickness is ascertained, and it admits of considerable variation, as above stated, it can only be maintained by firing small charges at frequent intervals; in other words, by firing the coal just as fast as it is consumed, so as to preserve an uniform thickness on the grate. This is a very severe duty for the firemen, and few are equal to it. A mechanical system of firing is, consequently, essential to maximum economic effect at minimum cost of labor. There being twice the chemically required quantity of oxygen with the coal as habitually fired in the furnaces of steam boilers (indisputably proved by chemical analyses), the combustion is sensibly perfect there, and as such perfection cannot be had with less than that proportion of oxygen, improvement on this original method ap-

pears to be hopeless; the plain grate is the simplest, the cheapest, the most convenient, the most efficient and the most economical.

The general truth of the above principles and deductions, is proven by the experiments whose data and results are now to be given, made on a number of boilers supplying steam to the factory of Messrs. Zuber, Reider & Co., at the Isle Napoleon, Alsace, to determine the economy of the Orvis apparatus, which injects mingled steam and air into the furnaces above the incandescent fuel on the grates. These experiments were undertaken for the purpose of ascertaining what royalty was to be paid for the use of the apparatus, as the amount was conditioned on the saving of coal it might effect. They were important to the company as regards its liability, and of the highest importance to the inventor, for a failure destroyed the entire future value of his patent. The greatest care was exercised in making the experiments, and the quantities obtained may be accepted as exact as they could possiby be measured.

The boilers were of the type known as "Chaudières des bouilleurs" in Europe, and as "Elephant boilers" in America, being composed of a large horizontal cylinder (the "chaudière") with two or three much smaller horizontal cylinders (the "bouilleurs") lying beneath the large one and connected with it by short pipes. The axes of all the cylinders are parallel to each other. The lower half of the "chaudière" is filled with water and is heating surface, the upper half is filled with steam. The "bouilleurs" are wholly filled with water and all of their surface is heating surface. The feed water coming from the heater is pumped into one end of one of the "boilleurs." The "chaudière" and its "bouilleurs" are set in brick masonry, which forms the furnace and the conduits or flues for the hot gases. The latter act first on the "bouilleurs" and afterwards on the lower half of the "chaudière." There were six of these boilers combined in one group. There was placed in the flue between the boilers and the chimney a heater into which the feed water was first pumped, passing through it into one of the "bouilleurs."

The aggregate grate surface of the six boilers was 149 square feet, and the aggregate water-heating surface was 2,763 square

feet, the ratio of the grate to the water-heating surface being 1 to 18.5436. The number of square feet of heating surface in the heater is unknown, but was probably sufficient, judging from the proportions generally employed, to make the ratio of the total water-heating surface in boiler and heater about 31 to 1 of grate surface.

The fuel was the bituminous coal of Saarebruck, in Alsace, and a special chemical analysis was made of its composition, the specimen being taken from the pile used during the experiments.

Specimens of the gases of combustion were drawn from the chimney at regular intervals during all the experiments and chemically analyzed. No combustible gas was at any time found in them. The mean of all these analyses during all the experiments was for the dry gases: carbonic acid $7\frac{8}{10}$ per centum, free oxygen 10.933 per centum, and nitrogen, by difference, 81.4 per centum.

The temperature of the air in the boiler room averaged during the experiments 70 degrees Fahrenheit, from which the variation was but little.

The temperature of the gases of combustion entering the chimney, taken between it and the heater, was noted at regular intervals.

The temperature of the feed water was observed at regular intervals, and is its temperature before entering the heater, the latter being in fact properly a portion of the boiler.

The feed water was measured in tanks. The coal and its refuse were carefully weighed. The pressure in the boilers was noted at regular intervals.

The data and results of the experiments will be found in the accompanying table, each column of which contains one experiment, except the last three columns, which contain the aggregate data and results of each series of experiments. For reference, each column is designated by a letter, and the lines containing the quantities are numbered. The latter are so fully described in the table as not to require further explanation.

The experiments consisted of three series, namely: 1st, without the Orvis apparatus; 2d, with the Orvis apparatus in con-

tinuous use; and 3d, with the Orvis apparatus in use for one-half of the time, the steam being shut off from it during the other half. The furnaces were charged at uniform intervals of time, and the time that the steam was used in the 3d series was from each charging for one-half the time to the succeeding charging.

The 1st series, without the Orvis apparatus, consists of experiments *A*, *B*, *C* and *D*. Of these, *A* and *B* were made with the six boilers in use, while *C* and *D* were made with only one boiler in use. The totals and means of the quantities belonging to these four experiments are given in column *K*, and will be used in the succeeding comparisons as representing the performance of the boilers without the Orvis apparatus.

The 2d series, with the Orvis apparatus in continuous use, consists of experiments *E*, *F* and *G*. Of these, *E* and *F* were made with the six boilers in use, while *G* was made with only one boiler in use. The totals and means of the quantities belonging to these three experiments are given in column *L*, and will be used in the succeeding comparisons as representing the performance of the boilers with the Orvis apparatus in continuous use.

The 3d series, with the Orvis apparatus in use intermittingly for one-half the time, consists of experiments *H*, *I* and *J*. Of these, *H* and *I* were made with the six boilers in use, while *J* was made with only one boiler in use. The totals and means of the quantities belonging to these three experiments are given in column *M*, and will be used in the succeeding comparisons as representing the performance of the boilers with the Orvis apparatus in intermittent use.

When the Orvis apparatus was used in experiments *E* and *F*, it was supplied with steam from one of the six boilers, but the quantity of feed water pumped into that boiler was not separately measured, this water being measured in common with the feed supply for the five other boilers. The steam from the one boiler, however, was not in common with the steam from the other five boilers. All the steam generated in the one boiler was used in the Orvis apparatus.

In experiments *H* and *I* the Orvis apparatus was similarly sup-

plied with steam from one of the six boilers, but for only one-half of the time, no steam being used in that apparatus during the remaining half of the time. When the apparatus was not in use the steam from the one boiler was used in common with the steam from the other five boilers. The feed-water supply to the one boiler furnishing the Orvis apparatus with steam was measured, as before described, in common with the feed water to the other five boilers.

After the experiments *A*, *B*, *E*, *F*, *H* and *I* were completed, experiments *C* and *D* were made with one boiler without the Orvis apparatus, but under the rate of combustion employed with that boiler during the experiments *E*, *F*, *H* and *I*, and the feed water was measured. Experiments *G* and *J* were also made with one boiler, and with the Orvis apparatus in continuous use during experiment *G*, and in intermittent use during experiment *J*, with the rate of combustion and with the proportional amount of steam supply to the apparatus that were used in experiments *E* and *F*, and *H* and *I*, respectively. The mean of the four determinations (experiments *C*, *D*, *G* and *J*) thus obtained of the weight of feed water used to supply the Orvis apparatus with steam during experiments *E* and *F* will be taken as the weight of steam expended for that purpose. In experiments *H* and *I* just half this weight of steam was expended by the apparatus.

During experiments *E*, *F*, *H* and *I*, the coal consumed in the one boiler supplying the Orvis apparatus with steam was weighed separately from the coal consumed in the remaining five boilers.

All the experiments were made of exactly the same duration, namely 12 hours, an important condition in securing *comparative* accuracy.

Each experiment was commenced with full fires, and the whole system in uniform operation, and was ended with the fires in the same condition and with the steam pressure and water level the same as at the commencement.

The total vaporization in the boilers, lines 16 and 17 of the table, is given from two assumed temperatures of feed water, namely, 100 and 212 degrees Fahrenheit, and is what it would have been had it been effected from these temperatures under

the standard atmospheric pressure and *with the experimental quantity of heat put into the feed water;* but, instead of these conditions, the vaporization took place under the boiler pressure, lines 7 and 8. Had the vaporizations been made under the atmospheric pressure from the temperatures 100 and 212 degrees, more heat would have been imparted to the water than the experimental quantity, because the temperature of the water in the boiler being then only 212 degrees instead of the temperature due to the pressure of the boiler steam, line 11, the difference between the temperature of the gases of combustion and the temperature of the boiler water would have been greater, and correspondingly more heat would have been taken by the latter out of the former. This effect would, of course, have increased the economic vaporization on lines 18 to 21, both inclusive, of the table; but, as the boiler pressure in colums *K*, *L* and *M* vary but very slightly, these causes do not affect the accuracy of the comparisons, nor of the absolute determinations to be hereinafter made.

The economic vaporization of water used in the comparisons is the one on line 21 of the table, being the pounds of feed water vaporized from the temperature 212 degrees under the atmospheric pressure, by the *experimental* quantity of heat put into it per pound of combustible, the "combustible" being what remains of the coal after deduction of its refuse; the "combustible," therefore, contains the hygroscopic water of the coal.

As regards the fumivority of the Orvis apparatus, or its smoke burning quality, the results were as follows: The time between charging the furnaces in experiments *A* and *B* without the apparatus, and in experiments *E* and *F* with it, was 6 minutes and 4 seconds. The time from the charging to the appearance of smoke at the top of the chimney was the same in all the experiments, and about 45 seconds.

The duration of black smoke with experiments *A* and *B* was 1 minute and 28 seconds, of brown smoke 1 minute and 10 seconds, and of light-colored smoke 2 minutes and 38 seconds.

With experiments *E* and *F*, there was neither black smoke nor brown smoke; there was light-colored smoke during 1

minute and 40 seconds, and very light-colored smoke during 3 minutes and 30 seconds.

With experiments *H* and *I*, the results were exactly the same as during experiments *E* and *F*, showing that as regards the prevention of smoke, the injection of the steam and air was only needed for the first half of the time between successive charges.

As regards the prevention of smoke, therefore, the Orvis apparatus may be considered quite successful.

The soot forming the black smoke was not sufficient to appear sensibly in the specimens analyzed of the gases of combustion, and therefore could not have appreciably influenced the economic vaporization.

The coal consumed was the Saarebruck coal of Alsace, Germany. It gave 63.65 per centum of coke. One pound of the coal produced by Berthier's method 24 pounds of metallic lead from litharge. The coal had, by special analysis, the following composition by weight, centessimally, exclusive of its ash and hygrometric water:

| | |
|---|---|
| Carbon, | 80.4790 |
| Hydrogen, | 6.9461 |
| Oxygen, | 9.8204 |
| Nitrogen, | 1.4371 |
| Sulphur, | 1.3174 |
| | 100.0000 |

The mean of Scheurer-Kestner's calorimetrical experiments on Saarebruck coal gave, for the heat of combustion of one pound of what remained after deducting its ash and hygroscopic water, 15,320.8 Fahrenheit units, deduction having been made for the latent heat of the steam produced by the combustion of the hydrogen of the coal, as there is no latent heat to be accounted for when the gases of combustion pass into the atmosphere at temperatures above 212 degrees Fahrenheit.

The coal consumed in the experiments contained 6 per centum of hygroscopic water. Now, the quantity denominated "combustible," on line 4 of the table, contains the hygroscopic water

of the coal, but not its ash; consequently, the chemical composition of this so-called "combustible" is, by weight, centessimally:

| | |
|---|---|
| Carbon, | 75.0838 |
| Hydrogen, | 6.4805 |
| Oxygen, | 9.1620 |
| Nitrogen, | 1.3408 |
| Sulphur, | 1.2290 |
| Water, | 6.7039 |
| | 100.0000 |

Neglecting, for the moment, the heat required to vaporize the 6.7039 per centum of hygroscopic water, the heat of combustion of the remaining (100.0000 — 6.7039 =) 93.2961 per centum of the "combustible" will be (15,320.8 × 0.932961 =) 14,293.708889 Fahrenheit units, deducting from which the heat, (.067039 × 1,108.578875 =) 74.318019 Fahrenheit units, required to vaporize this water under the atmospheric pressure from the atmospheric temperature (say 70 degrees Fahrenheit), there remains for the heat of combustion of one pound of the "combustible," on line 4 of the table (14,293.708889 — 74.318019 =) 14,219.390870 Fahrenheit units, which are equivalent to a vaporization per pound of "combustible" consumed of (14,219.390870 ÷ 965.7 =) 14.724439 pounds of water from the temperature 212 degrees Fahrenheit.

Experiments *C*, *D*, *G* and *J* of the table were made to determine the weight of feed water vaporized by one boiler during twelve hours for the conditions under which it was used during the experiments *E*, *F* and *G*, to supply with steam the Orvis apparatus in continuous action throughout these three experiments. The mean vaporization by the boiler for twelve hours was 11,151.525 pounds of water, which gives a total vaporization of (11,151.525 × $2\frac{1}{6}$ =) 24,161.6375 pounds, corresponding to the total vaporization 209,668.3 pounds, column *L*, line 6, of the table, during experiments *E*, *F* and *G*, in which the Orvis

apparatus was in continuous action. Hence the weight of steam injected into the furnace during experiments *E*, *F* and *G* was (24,161.6375 × 100 ÷ 209,668.3 =)11.523744 per centum of the feed water vaporized.

The volume of these 24,161.6375 pounds of steam under the atmospheric pressure to which that steam expands in the furnace, and at the chimney temperature of 487.7 degrees Fahrenheit, column *L*, line 10, is 866,054.41 cubic feet. In this expansion against the atmosphere (866,054.41 × 2,115. =) 1,831,705,077.15 foot pounds of work were done, equivalent to a transformation into it of (1,831,705,077.15 ÷ 789.25 =) 2,320,817.329427 Fahrenheit units of heat.

The sensible heat in this steam when entering the chimney, above what is due to its boiler temperature, is (24,161.6375 × 0.48 × 487.7 − 309.6 = 2,065,530.066600 Fahrenheit units.

The aggregate heat lost, so far as vaporization is concerned, by injecting the steam alone into the furnace, is then (2,320,817.329427 + 2,065,530.066600 =) 4,386,347.396027 Fahrenheit units, or (4,386,347.396027 ÷ 31,137=) 140.872512 units per pound of combustible, equivalent to ($\frac{140.872512 \times 100}{14219.39087}$ =) 0.990707 per centum of the heat of combustion. As there were utilized in the vaporization of water, column *L*, line 21, ($\frac{7.996847 \times 100}{14.724439}$ =) 54.310030 per centum of the heat of combustion, this vaporization would have been increased, had there been no loss of heat in the furnace by the steam injected there ($\frac{0.990707 \times 100}{54.31003}$ =) 1.824170 per centum, becoming (7.996,847 × 1.0182417 =) 8.142723 pounds of water from the temperature of 212 degrees Fahrenheit per pound of combustible.

Now, the mean vaporization of water from the temperature 212 degrees Fahrenheit when the Orvis apparatus was not in use (column *K*, line 21), was 8.464577 pounds per pound of "combustible;" there is, consequently, the heat in (8.464577 − 8.142723 =) 0.321854 pound of steam at the temperature of 212 degrees Fahrenheit to be accounted for per pound of "combustible." This heat was absorbed by the induced air injected into the furnace with the steam of the Orvis apparatus, and

amounted during experiments *E*, *F* and *G*, column *L* of the table, to (0.321854 × 965.7 × 31137. =) 9,677,828.215669 Fahrenheit units. There remains to ascertain in this connection the number of pounds of induced air injected into the furnace with the steam of the Orvis apparatus during experiments *E*, *F* and *G*, column *L* of the table.

Natural air at the temperature of 70 degrees Fahrenheit, and three-fourths saturated with aqueous vapor, has the specific heat 0.239784, consequently one pound of this air has in it when discharged into the chimney (1 × 0.239784 × 487.7 − 70.0 =) 100.157777 Fahrenheit units of heat.

One pound of such air has the volume 13.440822 cubic feet, which, after its expansion by the chimney temperature, becomes (13.440822 × 1.8497874 =) 24.862663 cubic feet, and in thus expanding against the atmosphere, does (24.862663 − 13.440822 × 2115 =) 24,157.193715 foot pounds of work, equivalent to a transmutation of (24,157.193715 ÷ 789.25 =) 30.607784 Fahrenheit units of heat.

Adding the immediately above two quantites together, there results (100.157777 + 30.607784=) 130.765561 Fahrenheit units of heat absorbed from the gases of combustion in the furnace by each pound of induced air, and as there were absorbed 9,677,828.215669 Fahrenheit units of heat by the total weight of induced air drawn into the furnace by the steam jets during experiments *E*, *F* and *G*, column *L* of the table, with the Orvis apparatus in continuous action, the division of that quantity by the 130.765561 gives the total number of pounds of induced air thus drawn into the furnace during these experiments, namely, 74,008.998559, which weight of air is, (74,008.998559 ÷ 24,161.6375=) 3.063079 times the weight of the injected steam. For each pound of "combustible" consumed the induced air injected into the furnace above the incandescent coal was (74,008.998559 ÷ 31,137 =) 2.376883 pounds.

As each pound of induced air absorbed 130.765561 Fahrenheit units of heat from the gases of combustion, and as there were 2.376883 pounds of air to each pound of combustible consumed, the heat absorbed by the induced air alone was

$\frac{130.765561 \times 2.376883 \times 100}{14.219.39087}$ =) 2.185849 per centum of the heat of combustion. And as it has been shown that 54.31003 per centum of the heat of combustion were utilized in the vaporization of water during experiments *E*, *F* and *G*, this vaporization would have been increased ($\frac{2.185849 \times 100}{54.31003}$ =) 4.024761 per centum. That is to say, the vaporization of water from the temperature of 212 degrees Fahrenheit, per pound of "combustible" during experiments *E*, *F* and *G*, as given on line 21 of the table, would have been, had there been no injection of steam and induced air into the furnace, 5.848931 per centum greater than it was. The direct comparison of this vaporization in column *K*, containing the mean of the experiments *A*, *B*, *C* and *D* made without the Orvis apparatus, with the corresponding vaporization in column *L*, containing the mean of the experiments *E*, *F* and *G* made with the Orvis apparatus in continuous action, gives, of course, the same result, namely, (8.464577 ÷ 7.996847=) 1.05848931. This loss is additional to the loss of the 11.523744 per centum of steam injected into the furnace, making the total commercial loss by the use of the Orvis apparatus (5.848931 + 11.523744=) 17.372675 per centum.

Experiments *H*, *I* and *J*, the means of which are given in column *M* of the table, were made with the Orvis apparatus in operation exactly one half of the time. The coal was fired in charges of uniform weight at uniform intervals of time, and the Orvis apparatus functioned from the throwing in of each charge for exactly half the time before charging again; for the remaining half of the time the steam was shut off from the Orvis apparatus and it did not function; therefore only one half as much steam and, consequently, one half as much induced air, were injected into the furnace with the experiments of column *M* as with those of column *L*. The economic vaporization, however, is not comparable with the economic vaporizations given in columns *K* and *L*, because the rate of combustion was considerably higher than in those cases. In the experiments of column *K* this rate, line 15, was 8.045224 pounds of "combustible" consumed per hour per square foot of grate surface, and in those of column *L* it was 8.117858 pounds, or very nearly the same; but,

in the experiments of column *M*, the rate of combustion was increased to 9.826300 pounds of "combustible" per square foot of grate surface, and the economic vaporizations are, therefore, not comparable. The economic vaporization in column *M* ought, with equal rate of combustion, to be midway between the economic vaporizations in columns *K* and *L*, that is to say it should have been $(\frac{8.464577 + 7.996847}{2} =)$ 8.230712 pounds of water per pound of "combustible" from the temperature of 212 degrees Fahrenheit. It really was 7.852383 pounds, hence, the inference is warranted that the increased rate of combustion diminished the economic vaporization of the "combustible" under the experimental conditions $(\frac{8.230712 - 7.852383 \times 100}{8.230712} =)$ 4.596552 per centum.

The combined weights of the constituents of one pound of the "combustible" portion of the coal, and of the oxygen with which they were combined during combustion, are given in the following table. The hygrometric water in the combustible, and the water resulting from the combustion of the hydrogen of the combustible, are given separately.

Each unit of weight of the carbon of the "combustible" combines with $2\frac{2}{3}$ times its weight of oxygen, the combined weights making $3\frac{2}{3}$ units of carbonic acid gas. Each unit of weight of the hydrogen of the "combustible" combines with eight times its weight of oxygen, the combined weights making nine units of water; and each unit of weight of the sulphur of the combustible combines with its own weight of oxygen, the combined weights making two units of sulphurous acid.

One pound of the "combustible" contains in fractions of a pound—

| | |
|---|---|
| Carbon, | 0.750838 |
| Hydrogen, | 0.064805 |
| Oxygen, | 0.091620 |
| Nitrogen, | 0.013408 |
| Sulphur, | 0.012290 |
| Hygroscopic water, | 0.067039 |
| Totals, | 1.000000 |

Weight in pounds of the gases of perfect combustion of one pound of the "combustible" portion of the coal—

| | |
|---|---|
| Carbonic acid, | 2.7530727 |
| Water of combination, | 0.5832450 |
| Nitrogen from air, | 10.6341627 |
| Nitrogen from combustible, | 0.0134080 |
| Sulphurous acid, | 0.0245800 |
| Hygroscopic water from combustible, | 0.0670390 |
| | 14.0755074 |

The aggregate weight of oxygen required for the above combinations is (2.002234667 + 0.51844 + 0.01229 =) 2.532964667 pounds, of which, supposing the oxygen (0.09162 pound) of the "combustible" to have entered into combination, (2.532964667 — 0.09162 =) 2.441344667 pounds were taken from the atmosphere, and with this weight of oxygen were associated 10.634-162690 pounds of atmospheric nitrogen.

The mean of the analyses of the dry gases of combustion drawn during experiments *A*, *B*, *C* and *D*, column *K* of the table, from the chimney, gave the following in volumes, centessimally:

| | |
|---|---|
| Carbonic acid, | 6.3 |
| Carbonic oxide, | 0.0 |
| Free oxygen, | 13.0 |
| Nitrogen by difference, | 80.7 |
| | 100.0 |

The above analysis shows that ($\frac{13}{20.96}$ = 0.620229, and 1.000000 — 0.620229 = 0.379771, which divided into unity gives) 2.633166 times as much oxygen were in the furnaces as was chemically required for perfect combustion.

The weight of oxygen, therefore, in the furnace per pound of "combustible" was (2.532964667 × 2.633166 =) 6.66971644 pounds, of which (6.66971644 — 0.09162 =) 6.57809644 pounds were drawn from the atmosphere, leaving (6.57809644 —

2.532964667 =) 4.04513177 pounds of unconsumed oxygen drawn from the atmosphere. To furnish this weight of oxygen (4.04513177 × 58.54638734 =) 230.827851448 cubic feet of atmospheric air at 70 degrees Fahrenheit, and three-fourths saturated with aqueous vapor, were required. As such air weighs 0.0744002 pound per cubic foot, the weight of it required to furnish the above weight of oxygen was 17.6200395 13 pounds per pound of "combustible." This weight was then the weight of the unchanged air in the furnace, and if there be added to it the weight of the gases of perfect combustion previously given, the sum will be (17.6200395 + 14.0755074 =) 31.6955469 pounds of gases in the furnace per pound of combustible.

The mean specific heat of the latter weight of gases is as follows:

| | Pounds of gases of combustion per pound of "combustible." | | Specific heat. | | Fahrenheit units of heat per degree temperature of gases. |
|---|---|---|---|---|---|
| Carbonic acid | 2.7530727 | × | 0.2169 | = | 0.597141469 |
| Water of combination | 0.5832450 | × | 0.4805 | = | 0.280249222 |
| Nitrogen from air | 10.6341627 | × | 0.2440 | = | 2.594735700 |
| Nitrogen from combustible | 0.0134080 | × | 0.2440 | = | 0.003271552 |
| Sulphurous acid | 0.0245800 | × | 0.1553 | = | 0.003817274 |
| Hygroscopic water from "combustible," | 1.0670390 | × | 0.4805 | = | 0.032212240 |
| Unchanged atmospheric air | 17.6200395 | × | 0.239784 | = | 4.225003554 |
| | 31.6955469 | | | | 7.736431011 |
| Mean specific heat | | | | | 0.244085740 |

The quantity of heat in the gases of combustion when they left the boiler can now be determined. The temperature of these gases in the chimney, column *K*, line 10, was 492.6 degrees Fahrenheit, deducting from which the atmospheric temperature of 70 degrees, there remain 422.6 degrees imparted by the heat of combustion, hence the gases of combustion, per pound of combustible, contained, when they left the boiler, (31.6955469 × 0.24408574 × 422.6 =) 3,269.415745249 Fahrenheit units of heat.

There still remains to determine the number of Fahrenheit units of heat per pound of "combustible" transmuted into the work of displacing the atmosphere by the difference of volume

between the volume of the gases of combustion when leaving the boiler at the temperature of the chimney and the volume of atmospheric air which entered the furnace at the temperature of 70 degrees.

It has been shown that the weight of oxygen drawn from the atmosphere per pound of combustible was 6.57809644 pounds. To furnish this weight of oxygen there would be required (6.57809644 × 58.54638734 =) 385.123782136 cubic feet of atmospheric air at 70 degrees Fahrenheit, and three-fourths saturated with aqueous vapor. This, then, was the volume of gases entering the furnace.

The pressure of the gases in the boiler from the mouth of the ash-pit to the top of the chimney is, of course, exactly the same, namely, that of the atmosphere, as their entire passage from the one locality to the other is made under the atmospheric pressure, although the density of the gases is very variable during that passage. The flow of the gases of combustion from ash-pit to chimney top is due to their lesser *density*, resulting from excess of temperature over that of the atmospheric gases.

The number of cubic feet of *dry* gases of combustion per pound of "combustible," escaping from the chimney is, when reduced to the temperature of 32 degrees Fahrenheit, as follows:

| | Pounds of dry gases of combustion per pound of combustible. | | Cubic feet of dry gases of combustion per pound of the same, at the temperature of 32 degrees. | | Cubic feet of dry gases of combustion per pound of "combustible" at the temperature of 32 degrees. |
|---|---|---|---|---|---|
| Carbonic acid ..... ..... | 2.7530727 | × | 8.1028658 | = | 22.307778626 |
| Nitrogen..... ..... | 10.6475707 | × | 12.7553607 | = | 135.813604857 |
| Sulphurous acid..... ..... | 0.02458 | × | 5.5137 | = | 0.135526746 |
| Unchanged air..... ..... | 17.6200395 | × | 12.389281867 | = | 218.299635873 |
| | | | | Total..... | 376.556546102 |

At the high temperature of the gases during their passage through the boiler, their coefficients of expansion would be sensibly the same, and equal to that of the perfect gases, hence, their volume, at the chimney temperature of 492.6 degrees, would be (376.556546102 × 1.937065 =) 729.414505777 cubic feet, to which volume must now be added the volume of the steam due to

the water of hydrogen combination and to the hygroscopic water of one pound of "combustible." The volume of one pound of steam under the atmospheric pressure is 26.31087 cubic feet, its temperature being 212 degrees; hence, the volume of the 0.650284 pound of steam when at the chimney temperature, and assuming for it the coefficient of expansion of a perfect gas, will be 26.671868350 cubic feet, which added to the above 729.414505777 cubic feet, gives for the bulk of the gases of combustion from one pound of "combustible" when emerging from the boiler into the atmosphere, 756.086374127 cubic feet. From this bulk there must now be subtracted the previously found 385.123782136 cubic feet of atmospheric air entering the furnace, leaving 370.962591991 cubic feet as the atmospheric volume displaced. This is equivalent to (370.962591991 × 2115 =) 784,585.882060965 foot pounds of work, which divided by 789.25, the mechanical equivalent of heat, gives 994.090442903 Fahrenheit units as the quantity of heat in the gases of combustion that was transmuted into the work of atmospheric displacement. Of course, the temperature corresponding to this heat was lost by these gases. The transmuted heat in question was heat absolutely annihilated, and appears no more in the gases of combustion whose heating effects, such as vaporization of water, raising the temperature of bodies external to the boiler by radiation and conduction, and in the temperature of the gases leaving the chimney, are due to only what remains of the heat of combustion of the "combustible" after deducting the heat transmuted into the work of atmospheric displacement.

The following table shows the distribution of the heat of combustion of one pound of the "combustible" portion of the coal.

| | |
|---|---|
| Fahrenheit units of heat due to the combustion of one pound of combustible | 14,219.390870000 |
| Fahrenheit units of heat absorbed in vaporizing water | 8,174.242008900 |
| Fahrenheit units of heat in the gases escaping from the boiler | 3,269.415745249 |
| Fahrenheit units of heat transmuted into work of displacing the atmosphere | 994.090442903 |
| | 12,437.748197052 |
| Fahrenheit units of heat imparted by radiation and conduction to external bodies by difference | 1,781.642672948 |
| Total | 14,219.390870000 |

The above distribution of the heat of combustion, expressed in per centum of its 14,219.39087 Fahrenheit units, is as follows:

| | |
|---|---|
| Absorbed in vaporizing water | 57.48659 per centum. |
| In the gases escaping from boiler | 22.99265 per centum. |
| In the work of displacing the atmosphere | 6.99109 per centum. |
| Lost by radiation and conduction | 12.52967 per centum. |
| | 100.00000 |

The losses of heat by radiation and conduction are very much greater with boilers set in brick masonry, and having their heating surfaces enveloped by the gases of combustion, as these were, than with boilers consisting of an external iron shell containing the heating surfaces and protected by non-conducting covering. In the one case, the thin brick wall has on its inner side the temperature of the gases of combustion; while, in the other case, the radiating temperature is only that of the water in the boilers, besides which, the bricks are not as good non-conductors of heat as the clothing with which iron-shelled boilers are covered. The above experimental loss of heat by radiation, namely, one-eighth roundly, is about what might have been anticipated in this case, and is confirmed by other experiments on boilers similarly set in masonry. This loss is about double the corresponding loss from well covered iron-shelled boilers, and, taken in connection with the loss due to the inleakage of air in the case of boilers set in masonry, makes up the average inferiority found in practice of about 10 per centum given by the latter kind of boilers compared with the economic vaporization of iron-shelled boilers, other things being equal.

As the loss of heat in the escaping gases increases with the chimney temperature, and as the loss of heat by the work of displacing the atmosphere also increases with the chimney temperature, obviously the first consideration for economic vaporization is to reduce these gases to as low a temperature as practicable, which can only be done by either lessening the rate of combustion or by increasing the quantity of heating surface, either expedient requiring more boiler to produce the same quantity of steam in equal time. The higher the temperature of the atmos-

pheric gases entering the furnace the less will be the loss of heat in the chimney gases and in the work of displacing the atmosphere; hence, when the boiler is in a confined apartment or boiler room, as generally the case on board steamers, the confined air becomes much heated by radiation and conduction from the external surfaces of the boiler, and there is thus recovered, if this air feeds the furnaces, a part of the heat loss by such radiation and conduction. As the radiation is irrespective of rate of combustion, the heat loss by it will be relatively greater at lower rates than at higher rates. Also, the higher the temperature of the air in the boiler room, the less will be the heat loss by radiation.

The losses of heat in the chimney gases and in the work of displacing the atmosphere, are less with less weight of air admitted to the furnace. Now, for perfect combustion—that is, for the coal gases to become saturated with oxygen—the oxygen supply per pound of coal must be just double of what is chemically required; therefore, if the air supply be reduced below what gives this limit, a part of the fuel will escape perfect combustion, and there will be a corresponding loss of heat; nevertheless this loss may be less in some cases than what is due to the admission of the full excess of air; the adjustment of the balance depends on the absolute conditions of temperatures, etc., but in general the best economic results will be obtained with air restricted a little more or less below the double quantity, so that some carbonic oxide will be produced. These considerations explain why a large variation in the thickness of the bed of incandescent coal on the grates does not sensibly affect the economic vaporization. The great point is to have no holes in the bed through which air can enter the boiler *en masse* and its oxygen escape combination. No air should be admitted at the bridge wall, and but little, if any, at the furnace door, if maximum economy is desired. The admission of air at the front of a furnace above the incandescent fuel may lessen smoke and preserve the furnace door, but these results are generally obtained at the cost of economy. In the case of the experimental boiler, the oxygen excess was enormous, roundly, $2\frac{5}{8}$ times as much as was chemically required,

probably due to carrying too thin a fire on the grate. Perfect combustion was thus secured, but, doubtless, at considerable sacrifice of economy.

The temperature of the gases of combustion in the furnace can be calculated from the experimental data, provided that temperature happens to be below about 1,800 degrees Fahhrenheit, the temperature caused by the dissociation of the carbonic acid gas and steam resulting from the combustion. In such a case the temperature depends on the number of pounds of gases of combustion in the furnace per pound of "combustible" consumed, on the mean specific heat of these gases, and on what remains of the heat of combustion of the pound of "combustible" after deduction is made of the heat transmuted into the work of atmospheric displacement. It has been shown that the weight of gases of combustion in the furnace per pound of combustible was 31.6955469 pounds, and that their mean specific heat was 0.24408574; the product of these quantities is 7.736431011. It has also been shown that the heat of combustion of the combustible after deduction is made of the heat transmuted into the work of displacing the atmosphere is 12,464.623857777 Fahrenheit units of heat; consequently, $\frac{12.464.623857777}{7.736431011} + 70° =$ 1,681 degrees Fahrenheit, which is the furnace temperature in question.

This low temperature was due to the large excess of air admitted. When the oxygen admitted to the furnace is just double of the chemical quantity required the temperature of the gases in the furnace is very close to 1,800 degrees.

The 70 degrees above added is, of course, the temperature of the air entering the ash-pit.

The writer has tried the following experiment to ascertain the temperature of the gases in the furnaces of many types of boiler, with many kinds of coal, and with all rates of combustion, from the lowest by natural draft to the highest by "blowers," and he has never been able to melt a silver coin in any case, nay, not even to produce the slightest disturbance of the delicate reeding on its edge. The experiment was made by nearly doubling the coin and placing it erect upon an ordinary

brick, then having the furnace well charged with coal, the brick, with the coin upon it, was placed in the center, with its lower surface resting on the upper surface of the coal. The bent coin was supported on only the two points given by its curved form, so that the gases of combustion could have free access to every part except the two supporting points. The brick and the coin remained in the furnace until the latter required recharging. The brick when withdrawn was at a white heat and the coin was intact. The melting point of a silver coin is certainly not above 1,900 degrees, and was never reached.

There must here be discriminated between the temperature of the gases of combustion in the furnace and the temperature due to the radiant heat from the bed of incandescent coal on the grate. These are quite different quantities, the latter being much greater than the former.

From the foregoing analysis will be perceived that the heat of combustion of a coal can be determined on the large scale by means of an ordinary steam boiler, from a knowledge of the chemical composition of the coal, of the chemical composition of the gases of its combustion, of the number of units of heat imparted to the water within the boiler, and of the quantity of heat externally radiated from it. In the case of an iron shelled boiler isolated from the ground as much as possible and thoroughly clad with non-heat-conducting covering, the loss of heat by radiation can be ascertained by direct measurement. The temperatures of the air entering the furnace, and of the gases of combustion in the chimney, as well as the barometer pressure, would be required, and the steam should be generated under the atmospheric pressure so as to reduce the losses from leakage and radiation to the minimum. The boiler should have an extremely large ratio of heating to grate surface, so as to reduce as much as possible the temperature of the gases in the chimney. The water level should be large and the steam room high and abundant. The steam room should be so arranged that the water of liquefaction due to radiation from the steam surfaces could be separately drawn off and weighed. The coal should be burned with a moderately high rate of combustion, the necessary draft

being given by a "blower." All the arrangements should be such that the maximum quantity of heat could be applied to vaporization, and that the vaporization should be so great absolutely as to reduce the heat applied to other effects to the minimum. An experiment should be of not less than 24 consecutive hours' duration, and longer if practicable. The chimney should be provided with a tight damper to be closed during the firing of the fuel and the cleaning of the fires, in order to exclude the inrush of air on such occasions through the furnace door. A very slight correction would be required for the heat in the refuse falling into the ash-pit. The results of experiments on the large scale made in this manner would be more trustworthy than any laboratory trials by means of a calorimeter.

The whole of the experiments could be performed by Naval Engineers, except the analysis of the coal, samples of which would have to be sent to a capable chemist for that purpose. The analysis of the chimney gases could be made by means of the simple Orsat apparatus by engineers without chemical training, and all the rest of the work could be better done by them than by others. The writer, near the close of his long service as Chief of the Bureau of Steam Engineering, had organized such a set of experiments, and with accomplished assistants, to be made in the Washington Navy Yard on American coals, but was not permitted to execute the idea, although exhaustive preliminary experiments had been completed on commercial wrought and cast iron, copper and brass, to determine their heat-conducting power under boiler conditions at different temperatures, and when of different thicknesses; also, on the non-heat-conducting power of cow hair felt, which, at that time, was exclusively used for the covering of boilers, steam pipes, etc. The subject is one, however, obviously of National importance. Although our coal is the foundation of our industries and wealth, we do not know to this hour the economic values and the absolute heats of combustion of its different varieties.

The results of the experiments made on the Orvis apparatus with the experimental boiler can be summarized as follows:

1st. The chemical analyses of the gases of combustion in the

chimney show that the quantity of free oxygen there present when the apparatus was not used, was $2\frac{5}{8}$ times what was required for complete saturation of the coal constituents. The excess of oxygen was, therefore, $1\frac{5}{8}$ times the chemical quantity.

2d. As the Orvis apparatus was based on the assumption that the coal constituents were not completely burned, owing to want of oxygen and of proper mixing of it with them, both erroneous assumptions, as proved by the analyses of the gases of combustion and by the boiler experimental results, obviously nothing could be gained by the use of this apparatus, and just as obviously something must be lost.

3d. The Orvis apparatus introduces into the furnace above the incandescent fuel on its grate a large mass of injected steam and induced air, which mass carries off a quantity of heat represented by the product of its weight, mean specific heat, and difference of temperature on entering the furnace and on leaving the boiler. Additionally, the heat was lost that was transmuted into the work done by the mass in displacing the atmosphere, owing to difference of bulk on entering the furnace and on leaving the boiler. The question was really answered by the chemical analysis of the gases of combustion in the chimney, and the experiment was wholly needless. There must, indeed, be a very incomplete combustion of the coal constituents, to enable any economic gain to be made by the use of the Orvis or other similar apparatus, as such gain can only be the difference between the losses by the apparatus and the gains by any better combustion it may produce. The experiment showed that the economic vaporization with and without the Orvis apparatus (line 21 of the table) compared nearly as 1.06 without and 1.00 with the apparatus. The quantity of injected steam and induced air was excessive, and there was already in the furnace a great deal more free oxygen than necessary, so that the conditions were dead against the apparatus, and success was impossible. With less injected steam and induced air, and with some unsaturated coal constituents to be burned in the furnace, other conditions favoring also, such as very thick fires and very rapid combustion, this

kind of apparatus may come off evenly in an evaporation test, involving neither gain nor loss, when the cost of the injected steam is not counted. The introduction of air alone under the atmospheric pressure through small holes in the furnace door, or, additionally, through the furnace front, generally produces, economically, neither loss nor gain. It assists in ventilating the fire room at the place most needing ventilation, it preserves the furnace door, and it diminishes smoke. Per contra, it diminishes the rate of combustion, and the rate of steam production. The introduction of air, either *en masse* or through small holes, at the bridge wall, always operates some economic loss without giving any mechanical gain as a compensation.

4th. The experimental data taken in connection with Scheurer-Kestner's determination of the heat of combustion of the Saarebruck coal, allows the distribution of the heat of combustion to be ascertained quantitatively, and allows the quantity of heat lost by radiation to be determined, a quantity not otherwise ascertainable with a boiler set in brick masonry. The data also allows the heat lost by the injected steam and by the induced air to be determined separately, and enables the weight of air induced to be calculated, the weight of steam injected being known by measurement.

5th. In relation to the smoke prevention effected by the Orvis apparatus: That which in popular language is called "smoke," is excessively comminuted carbon precipitated by the decomposition by heat of a very small part of the hydro-carbon gases distilled from the coal in furnaces, the precipitate having escaped combustion for want of being brought into contact with oxygen. Even in exaggerated cases, the precipitate is but a very small fraction of the carbon of the crude coal, and when deposited in mass upon the flue surfaces is called "soot." There is present in the furnace, simultaneously with its soot, oxygen enough for its combustion and temperature enough for its ignition, all that is required is that the soot and the oxygen be brought together; what is wanted is something to produce a merely mechanical mingling. A metallic stirrer would answer well, better, perhaps, than anything else, did the high temperature allow its use, but

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS MADE IN 1884 ON THE BOILERS IN
OF THE ORVIS SYSTEM OF STEAM AND AIR INJECTION INTO THE FURNACE

| Numbers on Lines. | | With the Orvis Apparatus not | | |
|---|---|---|---|---|
| | | Six boilers in use. | | One boil |
| | | October 3. A | October 4. B | October 19. C |
| | **Total Quantities.** | | | |
| 1 | Duration of the experiments in hours | 12. | 12. | 12. |
| 2 | Total number of pounds of Saarebruck bituminous coal consumed | 17,096.8 | 18,298.4 | 1,997.4 |
| 3 | Total number of pounds of refuse from the coal | 2,493.4 | 3,135.0 | 412.3 |
| 4 | Total number of pounds of combustible consumed | 14,603.4 | 15,163.4 | 1,585.1 |
| 5 | Per centum of the coal in refuse | 14.5840 | 17.1326 | 20.6418 |
| 6 | Total number of pounds of feed water vaporized | 98,282.0 | 114,355.9 | 11,157.6 |
| | **Mean Pressures and Temperatures.** | | | |
| 7 | Steam pressure in boilers in pounds per square inch above atmosphere | 61.3124 | 61.8124 | 71.3124 |
| 8 | Steam pressure in boilers in pounds per square inch above zero | 76.0000 | 76.5000 | 86.0000 |
| 9 | Temperature in degrees Fahrenheit of the feed water entering the heater | 61.5 | 62.3 | 46.9 |
| 10 | Temperature in degrees Fahrenheit of the gases of combustion between the heater and chimney | 489.2 | 507.2 | 437.0 |
| 11 | Temperature in degrees Fahrenheit of the steam in the boilers | 308.3 | 308.8 | 316.9 |
| | **Rate of Combustion.** | | | |
| 12 | Pounds of coal consumed per hour | 1,424.733333 | 1,524.866666 | 166.450000 |
| 13 | Pounds of combustible consumed per hour | 1,216.950000 | 1,263.616666 | 132.091666 |
| 14 | Pounds of coal consumed per hour per square foot of grate surface | 9.561967 | 10.234005 | 6.702684 |
| 15 | Pounds of combustible consumed per hour per square foot of grate surface | 8.167450 | 8.480630 | 5.319128 |
| | **Vaporization.** | | | |
| 16 | Total number of pounds of feed water vaporized from 100 degrees Fahrenheit | 104,472.401029 | 121,700.626954 | 12,038.580296 |
| 17 | Total number of pounds of feed water vaporized from 212 degrees Fahrenheit | 116,679.022573 | 135,920.205333 | 13,445.176351 |
| 18 | Pounds of feed water vaporized from 100 degrees by one pound of coal | 6.110641 | 6.650889 | 6.027126 |
| 19 | Pounds of feed water vaporized from 100 degrees by one pound of combustible | 7.153978 | 8.025946 | 7.594839 |
| 20 | Pounds of feed water vaporized from 212 degrees by one pound of coal | 6.824612 | 7.427983 | 6.731339 |
| 21 | Pounds of feed water vaporized from 212 degrees by one pound of combustible | 7.989853 | 8.963702 | 8.482226 |
| | **Constituents by Volume of Chimney Gases.** | | | |
| 22 | Carbonic acid | 6.3 | | |
| 23 | Carbonic oxide | 0.0 | | |
| 24 | Free oxygen | 13.0 | | |
| 25 | Nitrogen and other gases, by difference | 80.7 | | |

IN THE FACTORY OF MESSRS. ZUBER, RIEDER AND COMPANY, ISLE NAPOLEON, ALSACE, TO DETERMINE TH
ACES FOR INCREASING THE ECONOMY OF THE FUEL AND FOR THE PREVENTION OF SMOKE.

| OT IN USE. | | WITH THE ORVIS APPARATUS IN CONTINUOUS USE. | | | WITH THE ORVIS APPARATUS IN INTERMITTENT USE—HALF THE TIME. | | | GENERAL RESULTS OF THE | |
|---|---|---|---|---|---|---|---|---|---|
| boiler in use. | | Six boilers in use. | | One boiler in use. | Six boilers in use. | | One boiler in use. | With the Orvis apparatus not in use. Means and totals of experiments A, B, C and D. | With the Orvis apparatus in continuous use. Means and totals of experiments E, F and G. |
| 29. | October 20. | October 1. | October 2. | November 21. | October 6. | October 7. | November 22. | K | L |
| | D | E | F | G | H | I | J | | |
| | | 12. | 12. | 12. | 12. | 12. | 12. | ... | ... |
| | 2,116.4 | 16,434.4 | 18,518.8 | 2,187.0 | 22,454.1 | 21,164.4 | 2,112.0 | 39,509.0 | 37,140.2 |
| | 443.1 | 2,614.7 | 2,927.7 | 460.8 | 3,829.4 | 4,003.6 | 456.4 | 6,483.8 | 6,003.2 |
| | 1,673.3 | 13,819.7 | 15,591.1 | 1,726.2 | 18,624.7 | 17,160.8 | 1,655.6 | 33,025.2 | 31,137.0 |
| 18 | 20.9365 | 15.9099 | 15.8093 | 21.0700 | 17.0543 | 18.9167 | 21.6070 | 16.4109 | 16.1636 |
| | 11,173.2 | 98,405.5 | 100,297.0 | 10,965.8 | 121,836.2 | 114,355.9 | 11,307.5 | 234,970.7 | 209,668.3 |
| 24 | 71.3124 | 63.3124 | 61.3124 | 71.3124 | 62.3124 | 62.3624 | 71.3124 | 62.5286 | 62.7548 |
| 00 | 86.0000 | 78.0000 | 76.0000 | 86.0000 | 77.0000 | 77.2500 | 86.0000 | 77.2162 | 77.4424 |
| | 44.0 | 63.1 | 62.0 | 42 8 | 57.6 | 56 8 | 44.2 | 59 4 | 61.4 |
| | 442 4 | 473.0 | 503.6 | 462.2 | 530 6 | 537.8 | 455.0 | 492.6 | 487.7 |
| | 316 9 | 310.1 | 308.3 | 316.9 | 309.2 | 309 4 | 316.9 | 309 4 | 309.6 |
| 0000 | 176 366666 | 1,369.533333 | 1,543.233333 | 182.250000 | 1,871.175000 | 1,763.700000 | 176.000000 | ... | ... |
| 1666 | 139.441666 | 1,151.641666 | 1,299.258333 | 143.850000 | 1,552.058333 | 1,430.066666 | 137.966666 | ... | ... |
| 2684 | 7.102014 | 9.291500 | 10.357271 | 7.338926 | 12.558220 | 11.836913 | 7.087248 | 9.624728 | 9.68297 |
| 9128 | 5.615100 | 7.729140 | 8.719854 | 5.792617 | 10.416499 | 9 597763 | 5.555705 | 8.045224 | 8.11765 |
| 0296 | 12,087.629390 | 104,507.366724 | 106,567.774592 | 11,873.245310 | 129,982.116777 | 121,033.475855 | 12,228.632997 | 250,299.237669 | 222,948.38662 |
| 6351 | 13,499.955667 | 116,718.073682 | 119,019.220912 | 13,260.522816 | 145,169.309676 | 135,175.103877 | 13,657.435493 | 279,544.359924 | 248,997.81741 |
| 7126 | 5.711411 | 6.359101 | 5.754573 | 5.429010 | 5.788792 | 5.718729 | 5.790072 | 6.335246 | 6.00288 |
| 4839 | 7.223827 | 7.562202 | 6.835167 | 6.878256 | 6.979018 | 7.052904 | 7.386224 | 7.579038 | 7.16024 |
| 7339 | 6.378735 | 7.102102 | 6.426940 | 6.063339 | 6.465159 | 6.386909 | 6.466589 | 7.075460 | 6.70426 |
| 2226 | 8.067863 | 8.445775 | 7.633792 | 7.681916 | 7.794451 | 7.876970 | 8.248239 | 8.464577 | 7.99684 |
| | | | 8.7 | | | 8.0 | | 6.3 | 8.7 |
| | | | 0.0 | | | 0 0 | | 0 0 | 0.0 |
| | | | 9.6 | | | 10.2 | | 13.0 | 9.6 |
| | | | 81.7 | | | 81.8 | | 80.7 | 81.7 |

ATA AND RESULTS OF THE EXPERIMENTS MADE IN 1884 ON THE BOILERS IN THE FACTORY OF MESSR
OF THE ORVIS SYSTEM OF STEAM AND AIR INJECTION INTO THE FURNACES FOR INCREASING THE

| | With the Orvis Apparatus not in Use. | | | | With the Orv |
|---|---|---|---|---|---|
| | Six boilers in use. | | One boiler in use. | | Six boilers |
| | October 3. A | October 4. B | October 19. C | October 20. D | October 1. E |
| **Total Quantities.** | | | | | |
| ours | 12. | 12. | 12. | 12. | 12. |
| ebruck bituminous coal consumed | 17,096.8 | 18,298.4 | 1,997.4 | 2,116.4 | 16,434.4 |
| e from the coal | 2,493.4 | 3,135.0 | 412.3 | 443.1 | 2,614.7 |
| ustible consumed | 14,603.4 | 15,163.4 | 1,585.1 | 1,673.3 | 13,819.7 |
| | 14.5840 | 17.1326 | 20.6418 | 20.9365 | 15.9099 |
| water vaporized | 98,282.0 | 114,355.9 | 11,157.6 | 11,175.2 | 98,405.5 |
| **an Pressures and Temperatures.** | | | | | |
| ds per square inch above atmosphere | 61.3124 | 61.8124 | 71.3124 | 71.3124 | 63.3124 |
| nds per square inch above zero | 76.0000 | 76.5000 | 86.0000 | 86.0000 | 78.0000 |
| eit of the feed water entering the heater | 61.5 | 62.3 | 46.9 | 44.0 | 63.1 |
| eit of the gases of combustion between the heater and chimney | 489.2 | 507.2 | 437.0 | 442 4 | 473.0 |
| eit of the steam in the boilers | 308.3 | 308.8 | 316.9 | 316 9 | 310.1 |
| **Rate of Combustion.** | | | | | |
| ur | 1,424.733333 | 1,524.866666 | 166.450000 | 176 366666 | 1,369.533333 |
| d per hour | 1,216.950000 | 1,263.616666 | 132.091666 | 139.441666 | 1,151.641666 |
| ur per square foot of grate surface | 9.561967 | 10.234005 | 6.702684 | 7.102014 | 9.191500 |
| d per hour per square foot of grate surface | 8.167450 | 8.480630 | 5.319128 | 5.615100 | 7.729140 |
| **Vaporization.** | | | | | |
| water vaporized from 100 degrees Fahrenheit | 104,472.401029 | 121,700.626954 | 12,038 580296 | 12,087.629390 | 104,507.366724 |
| water vaporized from 212 degrees Fahrenheit | 116,679.022573 | 135,920.205333 | 13,445.176351 | 13,499.955667 | 116,718.073682 |
| from 100 degrees by one pound of coal | 6.110641 | 6.650889 | 6 027126 | 5.711411 | 6.359101 |
| from 100 degrees by one pound of combustible | 7.153978 | 8.025946 | 7.594839 | 7.223827 | 7.562202 |
| from 212 degrees by one pound of coal | 6.824612 | 7.427983 | 6.731339 | 6.378735 | 7.102102 |
| from 212 degrees by one pound of combustible | 7.989853 | 8.963702 | 8.482226 | 8.067863 | 8.445775 |
| **uents by Volume of Chimney Gases.** | | | | | |
| | 6.3 | | | | 8.7 |
| | 0.0 | | | | 0.0 |
| | 13.0 | | | | 9.6 |
| ference | 80.7 | | | | 81.7 |

ESSRS. ZUBER, RIEDER AND COMPANY, ISLE NAPOLEON, ALSACE, TO DETERMINE THE EFFICIENCY HE ECONOMY OF THE FUEL AND FOR THE PREVENTION OF SMOKE.

| With the Orvis Apparatus in Continuous Use. | | | With the Orvis Apparatus in Intermittent use—Half the Time. | | | General Results of the Three Series. | | |
|---|---|---|---|---|---|---|---|---|
| ilers in use. | | One boiler in use. | Six boilers in use. | | One boiler in use. | With the Orvis apparatus not in use. Means and totals of experiments A, B, C and D. | With the Orvis apparatus in continuous use. Means and totals of experiments E, F and G. | With the Orvis apparatus in intermittent use. Means and totals of experiments H, I and J. |
| | October 2. | November 21. | October 6. | October 7. | November 22. | K | L | M |
| | F | G | H | I | J | | | |
| | 12. | 12. | 12. | 12. | 12. | ... | ... | ... |
| | 18,518.8 | 2,187.0 | 22,454.1 | 21,164.4 | 2,112.0 | 39,509.0 | 37,140.2 | 45,730.5 |
| | 2,927.7 | 460.8 | 3,829.4 | 4,003.6 | 456.4 | 6,483.8 | 6,003.2 | 8,289.4 |
| | 15,591.1 | 1,726.2 | 18,624.7 | 17,160.8 | 1,655.6 | 33,025.2 | 31,137.0 | 37,441.1 |
| | 15.8093 | 21.0700 | 17.0543 | 18.9167 | 21.6070 | 16.4109 | 16.1636 | 18.1266 |
| | 100,297.0 | 10,965.8 | 121,836.2 | 114,355.9 | 11,307.5 | 234,970.7 | 209,668.3 | 247,499.6 |
| | 61.3124 | 71.3124 | 62.3124 | 62.5624 | 71.3124 | 62.5286 | 62.7548 | 62.8250 |
| | 76.0000 | 86.0000 | 77.0000 | 77.2500 | 86.0000 | 77.2162 | 77.4424 | 77.5126 |
| | 62.0 | 42 8 | 57.6 | 56 8 | 44.2 | 59 4 | 61.4 | 56.6 |
| | 503.6 | 462.2 | 530 6 | 537.8 | 455.0 | 492.6 | 487.7 | 530.5 |
| | 308.3 | 316.9 | 309.2 | 309 4 | 316.9 | 309 4 | 309.6 | 309.7 |
| 33 | 1,543.233333 | 182.250000 | 1,871.175000 | 1,763.700000 | 176.000000 | ... | ... | ... |
| 56 | 1,299.258333 | 143.850000 | 1,552.058333 | 1,430.066666 | 137.966666 | ... | ... | ... |
| 00 | 10.357271 | 7.338926 | 12.558220 | 11.836913 | 7.087248 | 9.624728 | 9.682975 | 12.001822 |
| 40 | 8.719854 | 5.792617 | 10.416499 | 9 597763 | 5.555705 | 8.045224 | 8.117058 | 9.826300 |
| 24 | 106,567.774592 | 11,873.245310 | 129,982.116777 | 121,033.475855 | 12,228.632997 | 250,299.237669 | 222,948.386626 | 263,244.225629 |
| 32 | 119,019.220912 | 13,260.522816 | 145,169.309676 | 135,175.103877 | 13,657.435493 | 279,544.359924 | 248,997.817410 | 294,001.849046 |
| 01 | 5.754573 | 5.429010 | 5.788792 | 5.718729 | 5.790072 | 6.335246 | 6.002886 | 5.756426 |
| 92 | 6.835167 | 6.878256 | 6.979018 | 7.058904 | 7.386224 | 7.579038 | 7.160240 | 7.030889 |
| 92 | 6.426940 | 6.063339 | 6.465159 | 6.386909 | 6.466589 | 7.075460 | 6.704267 | 6.429010 |
| 75 | 7.633792 | 7.681916 | 7.794451 | 7.876970 | 8.248239 | 8.464577 | 7.996847 | 7.852383 |
| | 8.7 | | | 8.0 | | 6.3 | 8.7 | 8.0 |
| | 0.0 | | | 0 0 | | 0 0 | 0.0 | 0.0 |
| | 9.6 | | | 10.2 | | 13.0 | 9.6 | 10.2 |
| | 1.7 | | | 81.8 | | 80.7 | 81.7 | 81.8 |

as such an expedient is impracticable, recourse must be had to gaseous streams injected for the same purpose, and the result produced of "smoke burning" is irrespective of whether air alone is used, or steam alone, or a mixture of both, the action being not chemical but purely mechanical. A similar injection of nitrogen or any inert gas would produce just the same effect. By such means a real combustion of the smoke is accomplished, that is to say, the small quantity of finely divided carbon present in the furnace by precipitation is consumed there, and consequently does not appear at the chimney top. The same result is produced by what is known as "back draft" in a furnace, the effect of which is to violently shake or agitate the gases of combustion there, mixing them thoroughly, when the smoke, no matter how dense previously, disappears as if by magic. The quantity of soot or precipitated carbon produced in a furnace by a given weight of different coals, is probably independent of the proximate composition of the coal gases, and depends on the quantity of those gases that can be distilled from it in a retort, and on the rapidity of the distillation. In twice the volume of gas obviously twice as many particles of precipitated carbon should escape combustion, and if the gas be distilled and pass out of the furnace in half the time, four times as many should escape, but if the quantity of gases be small and their distillation slow and gradual *pari passu* with the consumption of the coal, as with anthracite, there will be no smoke under any circumstances.

# XVII.

# INDICATOR TESTS.

By Assistant Engineer F. H. Conant, U. S. Navy.

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The Government contracts for ships with power requirements, attended with heavy forfeits and bonuses, has resulted, by reason of the necessity for measuring the power with the greatest possible exactness, in increased care in the manufacture of indicators for Government use, and increased accuracy in their standardization.

Until within a few years, the scale mark on an indicator spring was accepted without question, but advancing knowledge has shown that it is no more possible to make an errorless indicator spring than an errorless chronometer. The departure from its nominal scale, even if great, is not a serious defect if the amount can be determined. Its determination, however, has proved an intricate practical problem, but a partial solution has at any rate been effected, and one after another, the factors of error have been eliminated, to a degree that is very encouraging.

The first step in this direction was made, I am informed, by Mr. A. M. Mattice, then P. A. Engineer, U. S. Navy, who took exception to the manufacturers' practice of testing the springs when cold, and insisted on a hot test of some springs that he was to use. In 1888 a primitive arrangement for testing indicators with steam pressure was set up at the New York Navy Yard; and from it the present elaborate plant has developed. The history of its development would make interesting reading, but would occupy too much space. The most important event, however, was the abandonment of the spring gauge in favor of the mercury column. It was found that the gauge spring, when kept under tension for a short time, becomes fatigued and gives a

higher reading for the same pressure than it did at first. The amount of fatigue depends upon the intensity of the tension and the time interval for which it is maintained. After being, for twenty minutes, under a pressure gradually increasing from zero to 200 pounds, readings 1.5 pounds higher than the original have been obtained with a standard gauge. A few minutes' rest restores the spring to its normal condition. This action of the spring gauge destroys its efficiency as an accurate measuring instrument for sustained pressures, a fact which has a bearing not only on indicator tests, but on almost all data involving pressure per gauge.

Much credit for the present high degree of excellence of the plant attaches to Assistant Engineer W. D. Weaver, U. S. Navy, who has been the most active agent in its improvement. The apparatus designed by him is shown on Plate I.

The manifold *A* is a cylinder five feet long and three inches in internal diameter, with six cocks, *B*, for attaching indicators. Steam is admitted through the valve *C* and released through the valve *D*. Communication between the manifold and mercury column is through the water pot *E*, in which the water is kept at a constant level, and the pipe *E*. The height of the mercury is shown by the index *G*, which is attached to a float in the column. As the scale is graduated for only 25 pounds, the index is shifted on the cord the length of the scale for every increment or decrement of that amount, and the gauge *H* is used as an aid in following the index. The manifold and water pot are carefully lagged to prevent condensation, and the column and mercury reservoir are enclosed in a funnel through which the air circulates and keeps all parts of the column at the same temperature.

The indicator cords are led to the pulleys *I* on the shaft *J*, and a movement of the shaft produces a corresponding movement of the drums. The arrangement for applying the pencils is shown in Figure 2, Plate I. A bracket, *a*, is secured by set screws to the indicator frame between the cylinder and drum. An arm on the front of the bracket carries a lever pivoted at *b*. The lower end of the lever is connected by the links *c* with the arm *d* on

the shaft *K*. The upper end of the lever is connected by the link *e* with the clamp *f* set in the handle bracket of the indicator. Attached to the other side of the clamp is the spring *g*, which draws the pencil to the drum. The tension of the spring is adjusted by the screw *h*.

To operate the mechanism, the handle *L* on the pencil shaft *K* is raised and draws the pencils about one-sixteenth of an inch away from the drums, where they are held by a detent fixed on the armature of the magnet *M*. The handle *N*, on the drum shaft *J*, is then raised and pulls the drums around about one-third of a turn, where they are held by a detent engaging with the sector *O*. The machine is now set. To trip it, the push button *P* is pressed, the magnet releases the pencil shaft, and the pencils are drawn to the drums. The arm *Q*, on the lower side of the pencil shaft, is drawn back by a spring and strikes a "kicker," which in turn strikes the detent that holds the drum shaft, and causes it to release the shaft. The drums are moved by their own springs, and a mark about three-quarters of an inch long is made on the card. The whole operation is performed in a small fraction of a second. By means of the toothed sector *O* on the drum shaft, the line is made to occur on the proper section of the card.

For pressures below the atmosphere, communication to the main column is closed by the valve *R* and opened to the vacuum column by cock *S*. The switch *T* is shifted to bring the button *P'* into circuit. The connection to the air pump is through the valve and pipe *U* and a chamber that has about twelve times the cubic capacity of the manifold. A vacuum in the manifold is destroyed by admitting steam through the valve *V*. The scale on the vacuum column is graduated in both pounds and inches, and has at the bottom an arrangement of sliding wedges operated by the lever *W*, for raising and lowering the pot, so as to keep the level of the mercury always at the zero of the scale. The small pot *X* in the pipe between manifold and vacuum column is a separator to prevent water getting into the column.

A drawing of the Packard vacuum pump, which is used, is given on Plate II.

It produces a vacuum approaching the barometer reading within three-tenths of an inch. It has three $4\frac{1}{2}$ by 6 inch cylinders placed in a cast-iron box that serves as a bed plate and water jacket. It is belted to turn up to 70 revolutions per minute. The clearance is reduced to a minimum, and some of the valves in the piston have a tail stem extending through a sufficient distance to strike the bottom of the cylinder and lift the valve as the piston approaches the lower end of its stroke.

The remarkable results obtained by the pump are mainly due to an ingenious device in connection with the non-return valve at the bottom of the cylinder. (See Plate III.) The piston rod, which for $3\frac{1}{2}$ inches from its lower end is bored to a diameter of $\frac{9}{32}$ inches, and for 6 inches further to a diameter $\frac{1}{8}$ inch less, slides telescopically over the valve rod, the diameter of which, except at the upper end, is slightly less than that of the smaller bore. At the upper end the valve rod is split and spread to a diameter slightly greater than that of the smaller bore. On the upstroke the valve is lifted and held up by the friction of the rod in the smaller bore until, just before the end of the stroke, the end of the rod passes into the larger bore, and the valve and rod drop by their own weight.

A test proceeds as follows: The indicator is carefully examined to see that the bearing surfaces of the cylinder and piston are clean, and that the motion is in adjustment and lubricated. Then, with the indicator in a vertical position, and without the spring, the motion is raised and allowed to fall by its own, and the weight of the piston, to see that there is not excessive friction in the motion or between the piston and cylinder. Then, with the pencil so adjusted that a ray of light can be seen between its point and the drum, the piston is moved back and forth by blowing into, and exhausting from, the indicator cylinder, the pencil being observed to see that it moves in a direction parallel to the surface of the drum. With the pencil in the same position, the drum is pulled around to see that it is properly centered and cylindrical.

The indicator is then put on the manifold and "harnessed." Steam has, in the meantime, been blown through the manifold

to warm it up and work off the water that has collected in the pipes. When the steam in the manifold comes through dry, the indicator is warmed up and the card put on the drum. The silicated or "metallic" cards are used, as they permit the use of a metallic point with which a very fine, distinct line can be obtained.

After "setting" the machine, the manifold is exhausted to about 2 pounds below the atmosphere for the purpose of pulling the piston down preparatory to beginning the *up* test. The manifold is then opened to the atmosphere and the *up* zero line is made. The manifold is closed again and steam admitted in sufficient quantity to cause the mercury to rise in the column at the rate of about 2½ feet per minute. At 5 or 10-pound intervals, depending upon the scale of the spring, lines are made. When the limit of the test is reached, the pressure is carried one interval higher, and after shifting the drum shaft to bring the *down* lines in a new position, is allowed to fall, and the down lines are made, the manifold being opened to make the *down* zero. In testing springs for pressure below the atmosphere, the test is commenced at the lower limit, and lines are made at 2½ pound intervals.

When three tests have been made, a vertical line intersecting each set of horizontal lines is traced with the indicator. The piston is then taken out and callipered to one ten thousandth of an inch while hot. The card is taken off and marked with the make and number of the indicator, the number and scale of the spring, the temperature of the mercury column, and the date of the test. The accompanying cut is a fac-simile of a test card.

The cards are measured under a powerful glass with a scale graduated to one-hundredths of an inch; and in measuring, the fractional part of a graduation is estimated in tenths. This brings the readings approximately to one-thousandths of an inch; and in this connection it may be well to state that it is useless to carry the calculations for power further than two places of decimals, as that is beyond the limit of accuracy in measurement. The point from which the measurements in each test are made, and which is assumed as the zero of the instru-

*Plate I.*

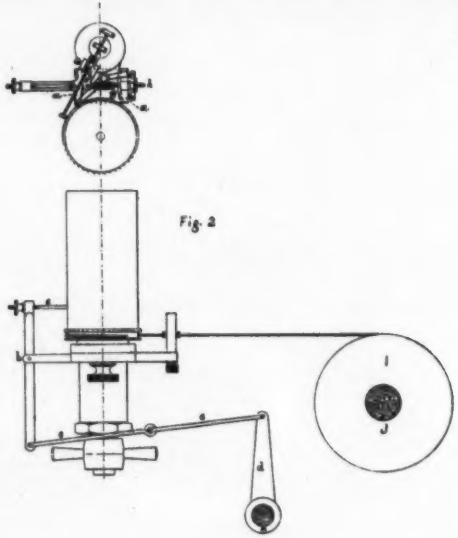
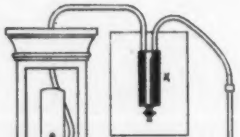


Fig. 2



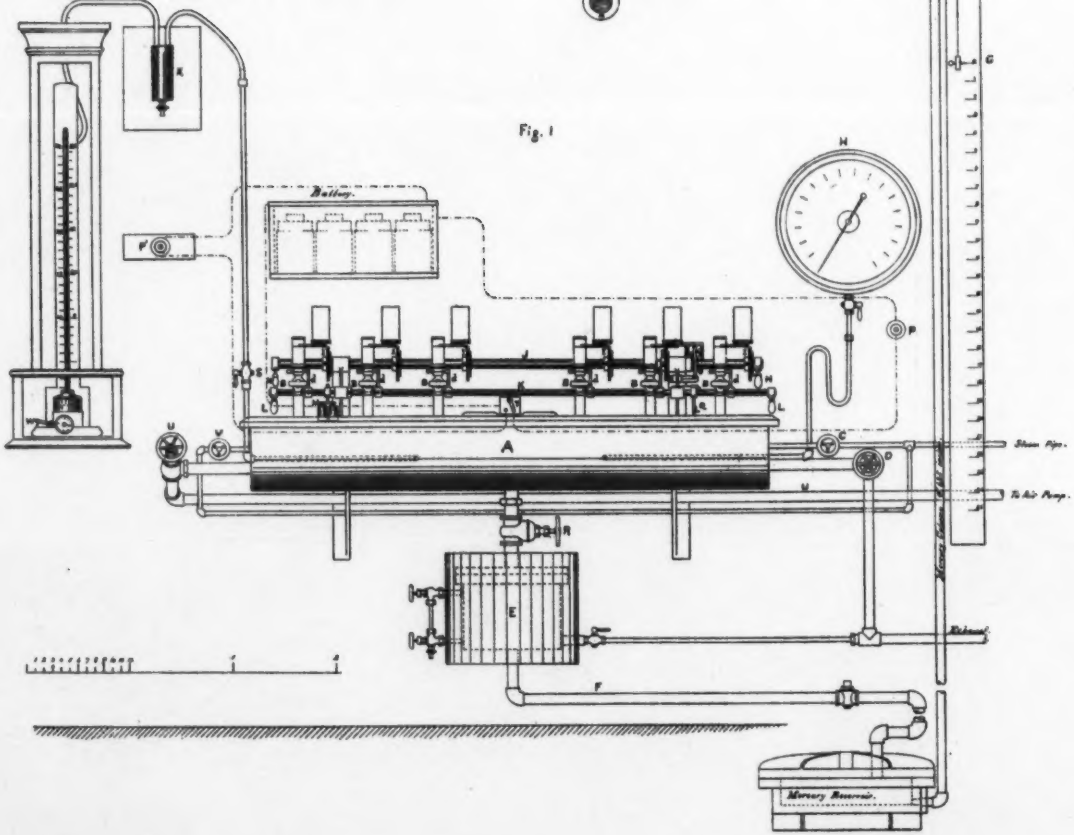
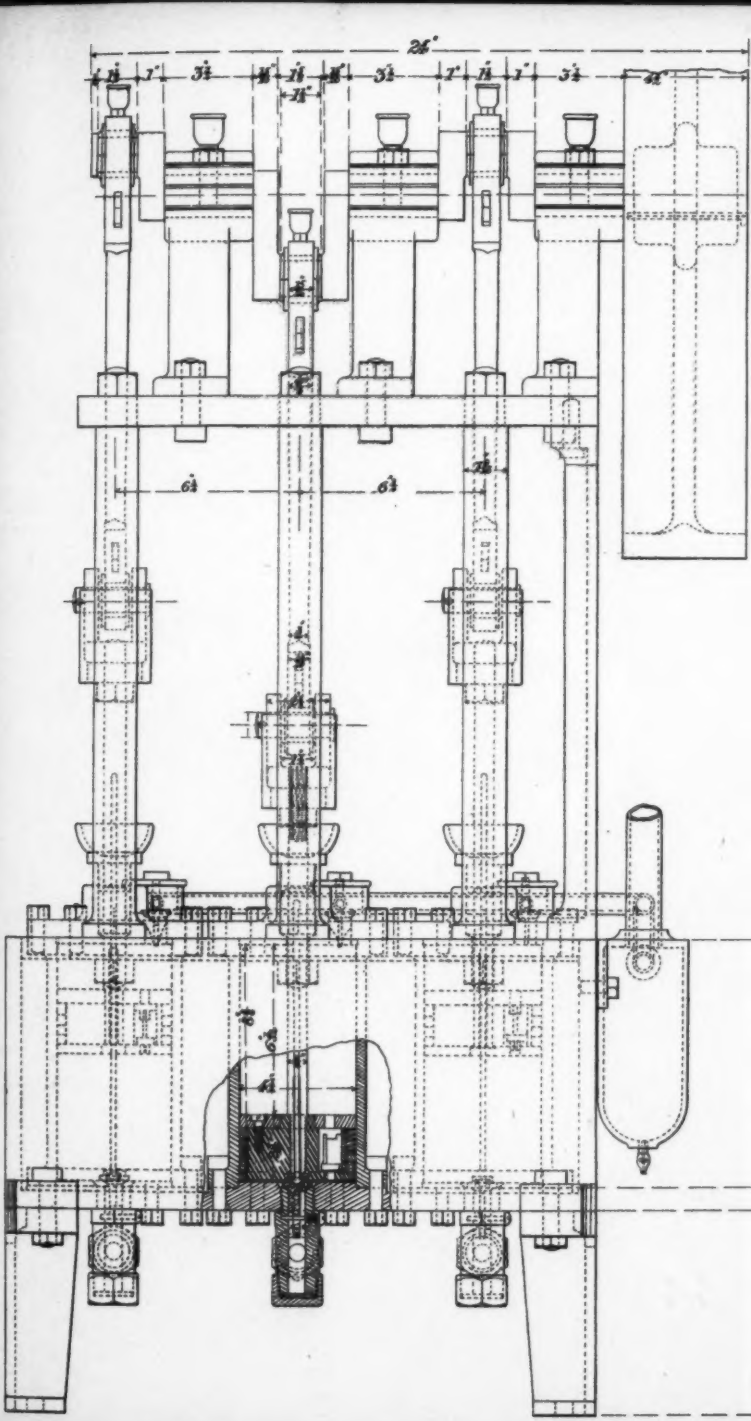
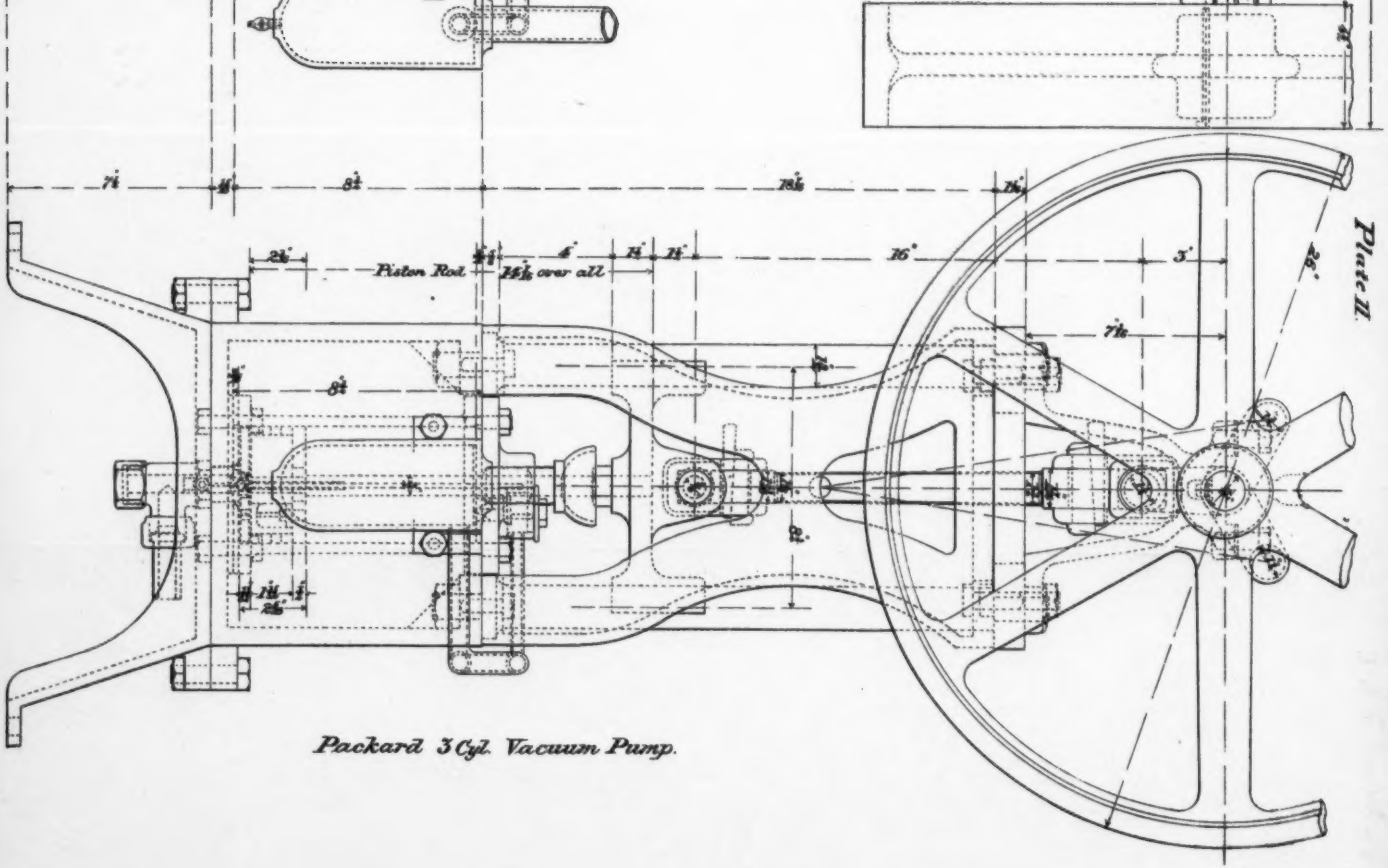


Fig. 1
Battery
A
E
F
G
H
K
L
M
P
R
S
U
V
W
Steam Pipe.
To Air Pump.
Mercury Reservoir.






*Packard 3 Cyl. Vacuum Pump.*

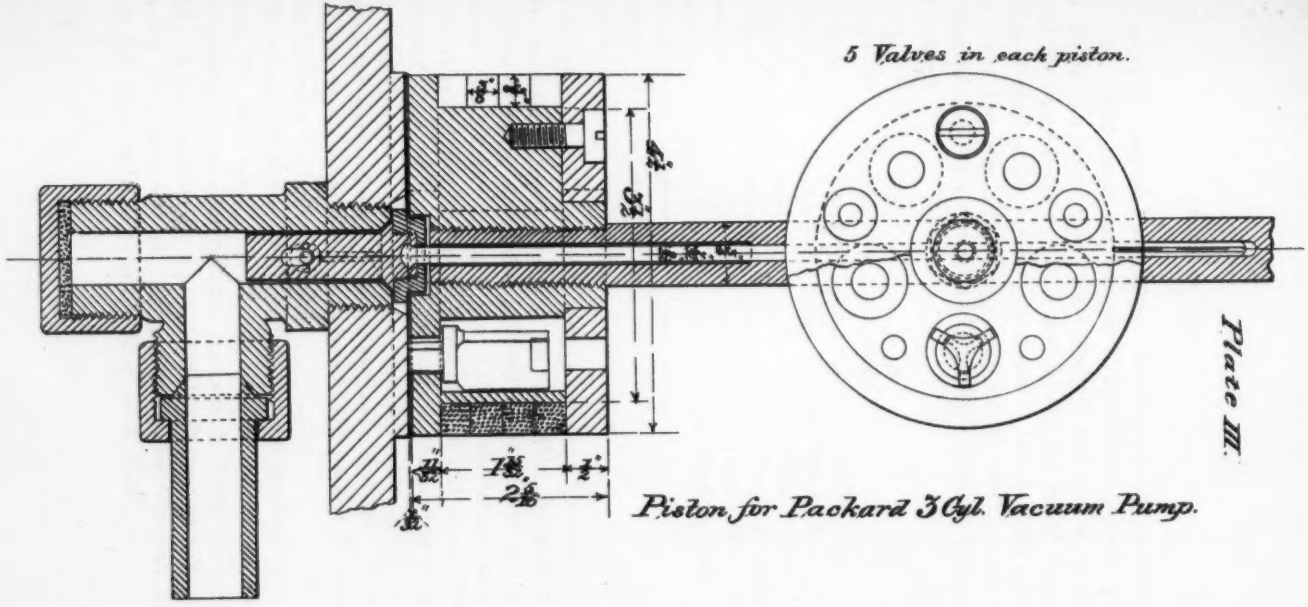
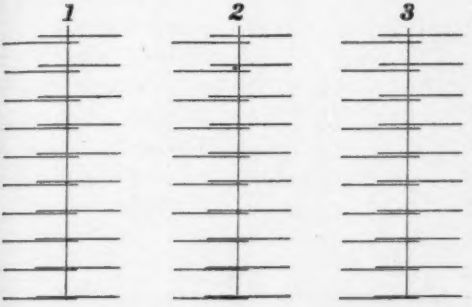



Plate III.

Piston for Packard 3 Cyl. Vacuum Pump.

ment, is on the vertical line midway between the up and down zero lines. The cards are measured by two observers, and, where there is a difference in their readings that cannot be reconciled, the mean is taken. The readings are corrected for depression of level in the mercury reservoir as the mercury enters the column, and also for the dilatation of the mercury due to any departure of the temperature from that for which the scale is graduated.




A criticism on this system of testing has been made by some of the indicator manufacturers, to the effect that the motion of the piston being so slow the conditions are not those of actual practice. That the criticism is just is not to be denied; but, as yet, no practical method of accomplishing a test under these conditions has been devised. At the same time, the makers may be met by their own claim that the indicators are equally well adapted for fast and slow running engines; and the question how slow can an engine be run before it will pervert the action of the indicator becomes pertinent.

The above description of the apparatus and method used in making the tests, which, it was suggested to the writer, would be of interest to the readers of the JOURNAL, he hopes to supplement, in a later number, by a paper which will contain a description and the results of some instructive experiments with indicators that have been made at the New York Yard.

## XVIII.

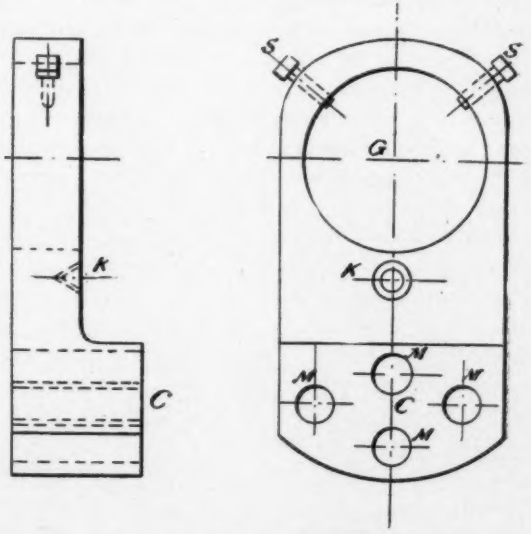
# THE FITTING UP OF A SOLID CRANK SHAFT.

By Passed Assistant Engineer J. A. B. Smith, U. S. Navy.

In an article by Assistant Engineer A. M. Hunt, U. S. Navy, on "Steel Crank Shaft Forgings" in the Journal, Vol. III, No. 1, page 5, this sentence occurs: "A rough-forged shaft, as it leaves the hammer, has considerable superfluous metal which must be reduced by machining." The first rough machining is done at the mill. The rough forging is placed on a surface plate, and gauged to see if it will finish up; if so, the webs are planed to about $\frac{1}{8}$ inch of the finished size, and then the superfluous metal is slotted out from between the webs. The couplings, journals, pins, and other parts are then rough turned in a manner similar to that of the finishing turning done at the machine shop and described below.

The section of rough-turned crank shaft having been received at the machine shop, is placed upon a surface plate, and, by means of an accurate surface gauge and from the dimensions given on the drawing, the centers of the shaft are established. Then the crank shaft is placed on these centers in a lathe, and from the dimensions given in the drawing the throw of the crank is proved, and special note taken of the amount of metal beyond the determined axis of the crank pin, so as to be sure that the pin will turn up to the proper size. If this and the other important dimensions are correct, the shaft journals, couplings, outside surfaces of webs, and crank pin edges of webs are turned to the finished size from the established centers of the shaft (these centers have been previously formed so that they are of the same taper as the centers of lathe on which the turning is done). Great care is taken to have the couplings on opposite ends of the crank shaft exactly alike in every dimension. This done, the shaft is removed to a planer and the webs are planed to the

finished thickness. It is then again placed upon the surface plate, and, working always from the established axis of the shaft, with the dimensions of the drawing as a guide, the height of the center of the crank pin is measured up from the surface plate by means of a surface gauge, and the position of the center of the crank pin determined. If it should happen that the center of the rough-turned pin is not at the same height above the surface plate as the axis of the shaft, the center of the pin must be shifted so as to divide equally the amount of metal on either side of the true axis of the pin. The true centers of the pin having been determined, the crank shaft, while still upon the surface plate, has fitted to it, around the couplings, two crank-pin-center-carrying discs, one on each coupling. These are of cast iron.




*G*, circular opening fitting over coupling on crank shaft; *SS*, set screws; *K*, hardened steel center, which is shaped to fit exactly the centers of the lathe in which the shaft is to be placed; *C*, counterbalance portion of disc; *MM*, holes that may be filled with lead to adjust the counterbalance.

The center *K* is laid off very carefully by accurate measurements on the surface plate.

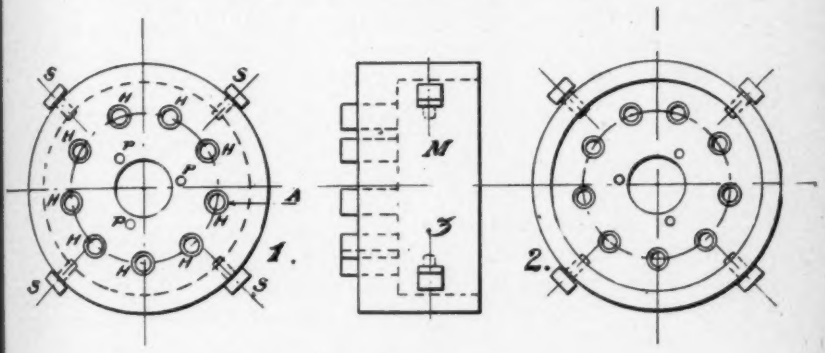
After the above discs are placed over the couplings, the centers *K* of the two discs are carefully measured with reference to the axis of the shaft, to see if they are directly in the prolonged axis of the crank pin. If the adjustment is correct, the set screws are set up tight, and the positions of the centers again proved. If any slight final adjustment is necessary, it is made by means of the set screws. The shafts and discs are then placed in the lathe, properly secured, and from the disc centers the crank pin is turned to finished size; also the inside faces of the crank webs and the heels of same. This done, the shaft and pin are bored in a horizontal boring mill. A hole large enough to admit of the passage of a small boring bar is first drilled through, and then the holes are bored in the usual manner. The crank shaft is then smoothed and the edges rounded by hand.

To lay off the holes in the shaft couplings so that the sections shall be reversible and interchangeable, it is necessary to decide first upon the proper number of holes. In a three-section crank shaft of the size required for *Cruisers 9, 10* and *11*, with cranks at 120°, nine holes is a good number.

Each section of crank shaft is laid in turn upon a surface plate with the axis of the shaft and pin in the same horizontal plane, and, using a surface gauge, a line is run from the center of the crank pin to the center of the shaft. On this line a distance from the center of the shaft toward the center of the crank pin, given by the drawing, is measured for the center of one coupling-bolt hole. This center is marked; then, with this distance as a radius and the center of the shaft as a center, a circle is described on the face of the coupling, and on this circle the centers of all holes must fall.

In order that all couplings may be exactly alike, and so drilled for bolts, a device called a "jig" is employed. 1 is the upper face; 2 is the under face; 3 is a side view. *H,H*, etc., are the holes with which the holes in each shaft coupling must correspond. They are fitted with hardened steel bushings. *P,P,P* are tapped holes for eye-bolts to raise the "jig" from the coup-

ling. *S,S,S* are set screws to secure "jig" to coupling. As stated above, the center of the bolt hole nearest the crank pin on each coupling is carefully determined. Then the crank shaft is up-ended under the drill press, the "jig" fitted over the coupling so that the center of the hole *A* is directly over the




marked center on the coupling, and using the holes *HH* as guides, the coupling-bolt holes are drilled through the coupling. Great care is bestowed upon the construction of the "jig." The holes must be *exactly* 40° apart (when nine coupling bolts are to be used), and their centers must be *exactly* the same distance from the center of the "jig;" which must, when placed on the coupling, be *exactly* in the prolonged axis of the shaft. The hardened steel bushings are designed to prevent wear of the holes by the drill which passes through them. Many trials of the "jig" holes as guides for drilling are made upon pieces of boiler iron, and it is not until the holes correspond in every position that the "jig" is considered fit to use.

All crank-shaft couplings and the forward couplings of forward sections of line shafts are drilled in this way, and the results attained at the Columbian Iron Works are eminently satisfactory.

It may be interesting to note, in connection with the subject of the fitting of crank shafts, a process which sometimes has to be employed. Passed Assistant Engineer A. C. Engard, U. S.

Navy, in his instructive article on the *Newark's* crank shafts in the JOURNAL, Vol. III, No. 1, page 12, says: "The crank shaft complete is now put in a lathe and the journals turned to size." It happened, in the case of the *Philadelphia's* crank shafts, that there was no lathe at the works large enough to machine the built-up shafts after the assembling of the parts, consequently the journals had to be turned to finished size before the building up.

The machining of the crank webs, in the case of the *Philadelphia*, differed from that of the *Newark* in that, in the former case, the holes in the webs for the shaft were bored to finished size in the boring mill, while the holes for crank pins were bored to within $\frac{1}{4}$ inch of the size given on the drawing; and, while this was proceeding, the shafts were being turned. The rest of the process of building up the crank shafts differs so materially from the process described by Mr. Engard, that it will be better, perhaps, to give it in detail.

The crank shafts are turned to finished size, the ends that are to be fitted to the webs being turned about $\frac{1}{100}$ inch larger than the holes in the crank webs. The key-ways are then laid off on the shafts from the drawing, so that when the cranks are in place they will be set at the proper angles with each other. Holes are drilled at the ends of the key-ways to clear the tool, and the key-ways slotted.

The crank webs and shafts are now ready to be put together. The web in each case is heated until it has expanded to the required size. The hole for the shaft is tried in several places by a tram made a little larger than the end of the shaft; and, when the required diameter has been reached, the web is put on a face-plate, the holes wiped out, and the shaft, which has been suspended from an overhead crane, is lowered in place, having a false key in key-way to guide it. The web is then gradually cooled, and, at the same time, adjusted to the fillet on the shaft. The face-plate prevents the shaft from going too far through the web. This is repeated until the crank webs are shrunk on all the sections of shaft. The keys are then fitted and driven home, and the ends cut off fair with the faces of the cranks. The keys

are made to bear lightly on their flat or widest side, and to a driving fit on the other sides. While this is going on, temporary bearings have been erected on a bed-plate in the shop. A long wooden straight-edge is placed through the engine bearings and their positions marked on it. From this straight-edge the temporary bearings are set. A wrought-iron collar about $3\frac{1}{2}$ inches wide is fitted to each end of these bearings, and these collars are fitted to the mandrel, thus avoiding the work of scraping the entire surface of the bearing. After the bearings have been secured on a bed-plate near a horizontal boring mill, the hollow cast-iron mandrel that has been used in fitting the regular engine bearings is placed in them, and a piano wire is stretched along and about 2 inches above it, and secured to upright pieces bolted to each end of the mandrel, to be used to detect any sag. After the bearings are adjusted, the mandrel is coated with red-lead paint, and revolved in them. It is then removed, and the collars are scraped to accurately fit the mandrel. This completed, the different sections of shafts with their cranks are placed in the bearings and the caps secured. The two webs at one end are placed at right angles to the bed-plate, and the boring bar, supported by a bearing at the outer end, is put through the holes for the crank pins, and the webs bored to the size called for in the drawing, the key-ways having been previously fitted with hard wood. The two webs at the other end can be bored at the same time by using a portable boring mill. When these two sets of crank webs have been bored to size (the eye of one web in each set being left a little large to allow the pin to slip in easily), the pins turned in the same manner as the shafts, and the key-ways cut, a cage made of perforated sheet iron, filled with shavings, wood, and charcoal, is placed around the ends of the webs.

A fire is started, and the holes in the webs for the pins are expanded to the required diameter. The fire is then removed, the holes wiped out, and the pin, having been previously suspended in a sling from an overhead crane, is carefully put in place. The pin is turned to the proper position in regard to the key-ways by a lever fitted between two pins that have been tapped into

one end for that purpose. The webs are then cooled by running water on them, gradually at first, care being taken to have them grip the pin at the same time. After the H.P. and L.P. pins have been shrunk into their respective webs, the shaft is turned to bring the I.P. webs in line and at right angles to the bed-plate, the boring bar is set, and the cranks shored. The I. P. webs are then bored out to the size required by the drawings, and the pin shrunk in, as described above for the H.P. and L.P. cranks. The shaft is then tested to see if pins and journals are in exact alignment. If so, the shaft is removed and set on blocks in the vicinity of the erected cylinders, the keys are fitted and driven home, and their ends cut off flush with the faces of the webs.

The eccentrics are then fitted and secured. The engine is erected in the shop, and the crank-shaft bearings having been scraped and fitted to the mandrel, the shaft is placed in its bearings, the caps secured, and the shaft revolved several times. It is then removed, and hard spots on the shaft journals are dressed off with a smooth file. It is then coated with red-lead paint, tried again, and hard-bearing places filed down. This is repeated until it shows a perfect bearing. Nothing is done to the bearings after they are once fitted to the mandrel, all the filing being done on the shaft journals.

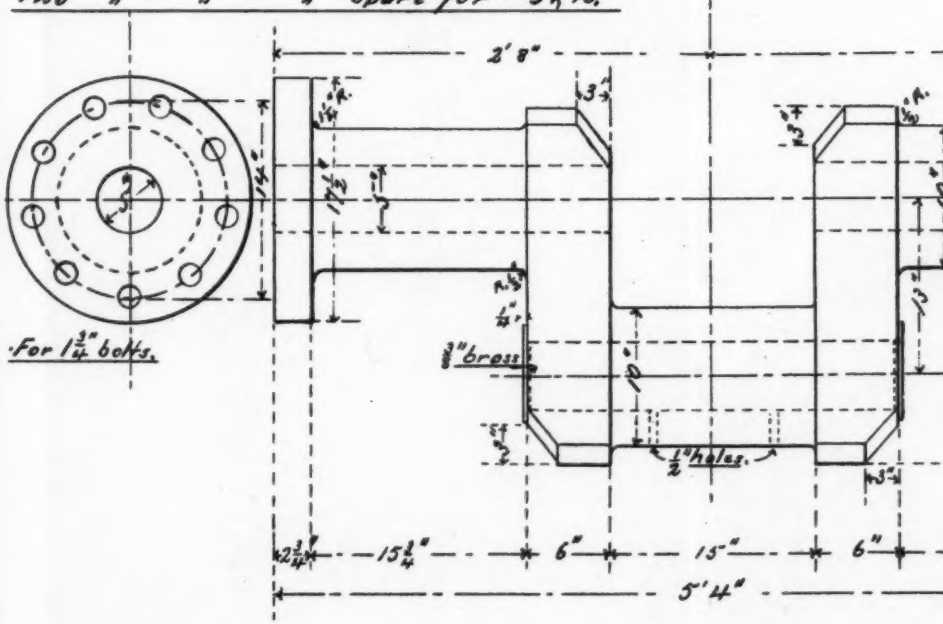
The *Philadelphia's* shaft had to be placed in position only three times before bringing it to a perfect bearing on all journals.

The author wishes to thank Assistant Engineers C. A. E. King and D. C. Redgrave, U. S. Navy, for assistance rendered in the preparation of this paper.

Six thus Cronk Shafts for engine #9.

" " " " " " #10,

Two " " " Spare for ##9 & 10.



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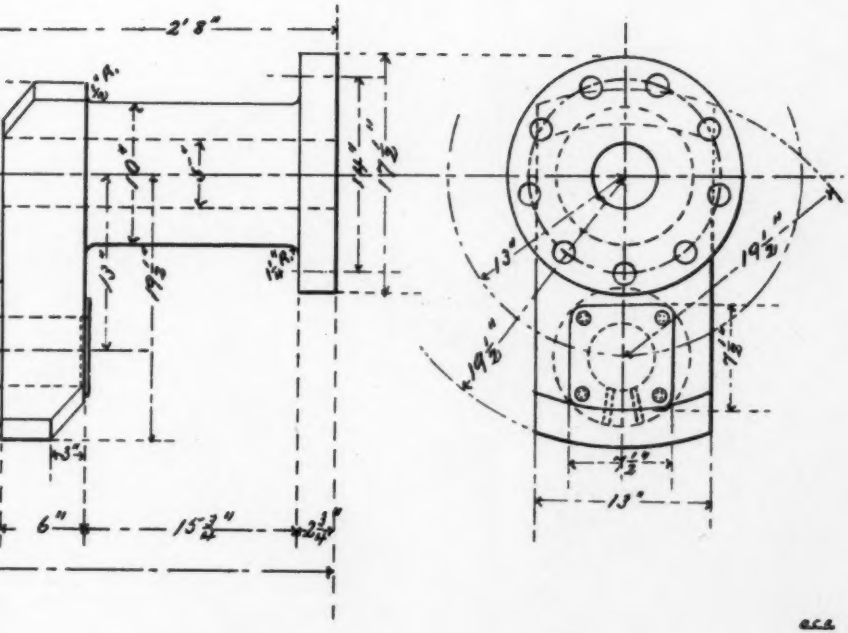


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# XIX.

# THEORY OF THE CENTRIFUGAL PUMP.

By Assistant Engineer H. G. Leopold, U. S. Navy.

## PART FIRST.

In preparing this paper on the centrifugal pump all available articles on the subject have been freely consulted and incorporated in the present article as far as they were considered valuable. The balancing of the wheel was suggested by a paper read before the Techniker Verein of San Francisco by J. Richards, which was kindly loaned to me by Mr. Ewald Schmidt. In the present article, however, the exact amount of that thrust is calculated. I must also thank Assistant Engineer Emil Theiss, U. S. Navy, for the translation of a paper from the German by Prof. G. Lindner.

Most of the formulæ given for the centrifugal pump consider only the surface velocity of the water at the tips of the vanes. Such formulæ are evidently incorrect, since the centrifugal pressure, or rather the difference of pressure between the entering and leaving edges of the wheel depends not only on the velocities of the water at these points, but also on the velocities at the intermediate points within the wheel, or, in other words, on the form of the vanes, and on the ratio of the radii at the entering and leaving edges of the wheel. Again, it will be noted that the form of the discharge conduit is not taken into consideration, whereas it is evident that the water must have a tangential velocity on leaving the tips of the vanes sufficient to balance the head against which the pump is working, and a radial velocity through the pump sufficient to give the required discharge. The first of these will be entirely wasted unless converted into pressure before being thrown into the discharge pipe, and in such a case the efficiency of the pump can not much exceed 50 per cent.

In the present paper only wheels having straight or spiral vanes and a constant radial velocity through the wheel will be considered. The discharge conduit will be supposed to have a constant breadth equal to that at the tips of the vanes, so that the radial velocity there will vary inversely as the radius.

Referring to Rankine's Applied Mechanics, article 618, for the general hydro-dynamic equations of liquid motion of invariable density, transforming them into cylindrical co-ordinates, and adapting them to steady motion, we shall have

$$X - \frac{dp}{\rho dx} = \frac{1}{g}\left(u\frac{du}{dx} + \frac{vdu}{dr} + \frac{wdu}{rd\theta}\right). \quad (1)$$

$$P - \frac{dp}{\rho dr} + \frac{w^2}{gr} = \frac{1}{g}\left(u\frac{dv}{dx} + \frac{vdv}{dr} + \frac{wdv}{rd\theta}\right). \quad (2)$$

$$\theta - \frac{1}{\rho}\frac{dp}{rd\theta} - \frac{wv}{gr} = \frac{1}{g}\left(u\frac{dw}{dx} + \frac{vdw}{dr} + \frac{wdw}{rd\theta}\right). \quad (3)$$

In the above equations $u$, $v$ and $w$ are the component velocities in the direction of $x$, $r$ and $\theta$ respectively; $X$, $P$ and $\theta$ the impulses impressed on the water in the same directions. The law of continuity for a space of invariable volume requires,

$$\left(\frac{du}{dx} + \frac{dv}{dr} + \frac{dw}{rd\theta}\right) = 0,$$

from which it is necessary, if the area through which the fluid passes is constant, that each of these quantities should be zero; and if that area varies, the velocity in that direction must vary inversely as that area. For the wheel under consideration, since those areas are constant, each of these reduces to zero, and for the discharge conduit assumed, $\frac{dv}{dr}$ must be retained and integrated.

It will be further noted that $w$ and $v$ vary with $r$ only, and the pressure, considering the vortex as a whole, will be considered to vary only with $r$ (the radius). Striking out these quantities, we shall have for the vortex within the wheel,

$$(1)\ X = \frac{vdu}{gdr}, \quad (2)\ \frac{dp}{\rho dr} = \frac{w^2}{gr}, \quad (3)\ \theta = \frac{wv}{gr} + \frac{vdw}{dr}.$$

These equations are the fundamental equations for solving any

problem connected with any form of centrifugal pump. When $v$ is not constant, however, equation (2) becomes

$$\frac{dp}{\rho dr} = \frac{w^2}{gr} + \frac{vdv}{dr}.$$

Equation (1) gives the impulse of the entering water against the disc of the wheel.

$$X = \frac{vdu}{gdr}, \text{ and } dr = vdt;$$

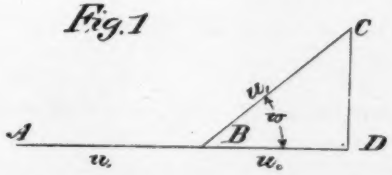
hence we have $X = \frac{1}{g}\int\frac{du}{dt} = \frac{1}{g}\left(u_1 - u_0\right)$.

In Fig. (1) let $AB = u_1$, $BD = u_0$, and let $BC$ = velocity through the wheel = velocity in suction pipe = $u_1$; then we shall have, letting $\omega$ = angle through which the water is deflected in passing through the wheel,

$BD = u_0 = u_1 \cos \omega$.

$$X = \frac{u_1}{g}\left(1 - \cos \omega\right), \text{ per pound of water.}$$

*Fig. 1*




$$X_{\text{total}} = \frac{\text{No. cu. ft. per sec.} \times \text{density}}{g} u_1\left(1 - \cos \omega\right)$$

$$= \frac{\rho \text{ Area of suction pipe}}{g} u_1^2\left(1 - \cos \omega\right) = \frac{\rho A u_1^2}{g}\left(1 - \cos \omega\right)$$

= Thrust due to entering water.

Equation (2) gives for the wheel,

$$\frac{dp}{\rho dr} = \frac{w^2}{gr}.$$

In the wheel at present under consideration the angular velocity of the water is assumed constant, a condition requiring

spiral or straight vanes. (This will be considered later on.) $w$ will then be equal to $\varphi r$, $\varphi$ being the angular velocity of the water and not of the wheel. Whence we have

$$\int_{p_0}^{p_1}\frac{dp}{\rho}=\int_{r_0}^{r_1}\frac{\varphi^2 r^2 dr}{gr}=\frac{1}{\rho}\left(p_1-p_0\right)=\frac{\varphi^2 r_1^2}{2g}\left(1-\frac{r_0^2}{r_1^2}\right). \qquad \left(h_1\right)$$

In this expression $r_1$ is the radius at the outer edges of the vanes, $r_0$ that at the inner edges.

This gives the difference of head existing between the tips of the blades; and this pressure is one of the principal elements in balancing the column of water in the discharge pipe, indeed in some cases all that is available.

The centrifugal pressure in the pump with wheels not encased tends to force the wheel endwise, and the end thrust due to it will be

$$t_2=\int_{r_0}^{r_1}p2\pi r dr=\frac{2\pi\rho}{2g}\int_{r_0}^{r_1}\varphi^2 r^3 dr=\frac{\rho\pi\varphi^2}{4g}\left(r_1^4-r_0^4\right)=$$

$$\frac{\rho\pi\varphi^2 r_1^2}{4g}\left(r_1^2-\frac{r_0^4}{r_1^2}\right)=t_2.$$

At the rear of the disc the pressure of the water is that due to the pressure at the tips of the vanes and is equal to $p_1=\frac{\rho\varphi^2 r_1^2}{2g}$, and the thrust due to it will be,

$$\text{Area of disc}\times p_1=t_3=\frac{\rho\varphi^2\pi r_1^4}{2g}.$$

The total end thrust of a wheel not encased will be,

$$T_t=\frac{\rho\varphi^2\pi r_1^4}{2g}-\frac{\rho\pi\varphi^2}{4g}\left(r_1^4-r_0^4\right)-\frac{\rho A u_1^2}{g}\left(1-\cos\omega\right)=$$

$$\frac{\pi\rho\varphi^2}{4g}\left(r_1^4-r_0^4\right)-\frac{\rho A u_1^2}{g}\left(1-\cos\omega\right).$$

This thrust will be in a direction opposite to that from which the water is coming. The wheel may be balanced, however, by

continuing the vanes behind the disc, and cutting holes through the disc at the center, as illustrated in Fig 4.

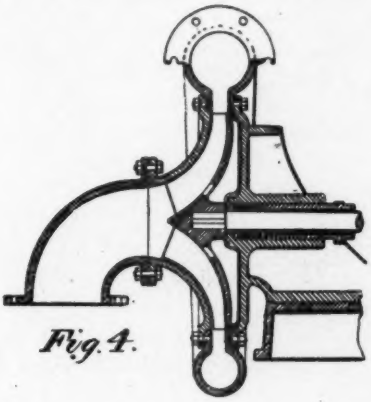


*Fig. 4.*

Equation (3) gives,

$$\theta = \frac{wv}{gr} + \frac{vdw}{gdr},$$

and the power required to drive the pump will be, in foot pounds per pound of water pumped:

Since $vdt = dr$ and $v = \frac{dr}{dt}$,

$$\int_{r_0}^{r_1} \theta\varphi r \frac{dr}{dt} = \int_{r_0}^{r_1} \frac{\varphi^2 r}{g}\, dr + \int_{w_0}^{w_1} \frac{w dw}{g}$$

$$= \frac{\varphi^2 r_1^2}{g}\left(1 - \frac{r_0^2}{2r_1^2}\right).$$

Let $Q$ = cu. ft. per sec.; then the foot pounds per second will be

$$F_p = \frac{\rho Q \varphi^2 r_1^2}{g}\left(1 - \frac{r_0^2}{2r_1^2}\right).$$

Beyond the tips of the vanes the water forms a free spiral vortex. In such a vortex the total energy of the water remains

constant, since it is acted on by no forces. (For the nature of such a vortex, see Rankine's Applied Mechanics.) Since the energy remains constant, we have, if $p_m$ = maximum pressure equivalent to the total head,

$$p_m = p + \frac{\rho w^2}{2g}, \text{ and from equation (2),}$$

$$\frac{1}{\rho}\frac{dp}{dr} = \frac{w^2}{gr}; \quad \therefore \frac{2(p_m - p)}{\rho} = \frac{w^2}{g}.$$

$$\frac{dp}{2(p_m - p)} = \frac{dr}{r}; \quad \text{Log}_e(cr) = -\tfrac{1}{2}\log_e(p_m - p),$$

$p_m - p = \frac{c}{r^2} = \frac{\rho w^2}{2g}$, and $w$ varies inversely as the radius.

This property of the free vortex enables us to utilize the entire head due to the tangential velocity of the water on leaving the tips of the vanes.

The water on entering the discharge pipe should have a combined velocity equal to the velocity of discharge, and if the casing of the discharge conduit is of constant breadth, as we have assumed, the radial velocity will vary inversely as the radius. If we let $v_1$ = radial velocity of the water at the tips of the vanes, and $\varphi r_1$ the tangential velocity at the same point, then these velocities at any other point in the discharge conduit will be $\frac{v_1 r_1}{r}$, and $\frac{\varphi r_1^2}{r}$, and the combined velocity will be $\sqrt{\frac{v_1^2 r_1^2}{r_2^2} + \frac{\varphi^2 r_1^4}{r_2^2}}$ $= v_1$ at point of delivery to discharge nozzle.

Then $r_2 = r_1\sqrt{1 + \frac{\varphi^2 r_1^2}{v_1^2}}$ = radial distance from center of the wheel to center line of discharge nozzle.

For the vortex in the discharge conduit equation (2) gives,

$$\frac{1}{\rho}\frac{dp}{dr} = \frac{w^2}{gr} + \frac{vdv}{gdr}. \quad \text{Here } w = \frac{\varphi r_1^2}{r}.$$

$$\int_{p_1}^{p_2} p = \int_{r_2}^{r_1} \frac{\varphi^2 r_1^4}{gr^3} dr + \int_{v_2}^{v_1} \frac{vdv}{g} = \frac{\varphi^2 r_1^2}{2g}\left(1 - \frac{r_1^2}{r_2^2}\right) + \frac{v_1^2 - v_2^2}{2g}. \quad \left(h_2\right)$$

The water has a head due to the velocity of discharge $= \frac{v_1^2}{2g}$.

The total head thus available to effect the velocity of discharge, overcome friction, shock and eddies, and balance the head between the suction and delivery is equal to

$$H_t = h_1 + h_2 + \frac{v^2}{2g} = \frac{\varphi^2 r_1^2}{2g}\left(1 - \frac{r_0^2}{r_1^2}\right) + \frac{\varphi^2 r_1^2}{2g}\left(1 - \frac{r_1^2}{r_2^2}\right) + \frac{v_1^2 - v_2^2}{2g} + \frac{v_1^2}{2g}.$$

From which, knowing the total head required, and the ratio of $r_0$, $r_1$ and $r_2$, and the velocity of discharge required, we can find the angular velocity.

The head against which the water can be pumped, including friction, with the velocity of discharge, $v_1$, will be

$$H_e = h_1 + h_2 = \frac{\varphi^2 r_1^2}{2g}\left\{\left(1 - \frac{r_0^2}{r_1^2}\right) + \left(1 - \frac{r_1^2}{r_2^2}\right)\right\} + \frac{v_1^2 - v_2^2}{2g}.$$

The efficiency will be, neglecting friction, $\frac{\rho Q H_e}{F_p} =$

$$Ef. = \frac{\frac{Q\rho\varphi^2 r_1^2}{2g}\left(2 - \frac{r_0^2 r_2^2 + r_1^4}{r_1^2 r_2^2}\right) + \frac{(v_1^2 - v_2^2)\rho Q}{2g}}{\frac{\rho Q \varphi^2 r_1^2}{g}\left(1 - \frac{r_0^2}{2r_1^2}\right)} =$$

$$\frac{\left(1 - \frac{r_0^2 r_2^2 + r_1^4}{2r_1^2 r_2^2}\right) + \frac{v_1^2}{2\varphi^2 r_1^2}\left(1 - \frac{r_1}{r_2}\right)}{1 - \frac{r_0^2}{2r_1^2}} = Ef.$$

We have now deduced all the formulæ necessary for solving any problem connected with this form of pump, but it remains to be shown what form of vane will give the water a constant angular velocity, $\varphi$, and the relation of the speed of wheel to that of the water.

Referring to Fig. 2, let $ar$ = velocity of the point $P$ on the vane, and let $v$ = radial velocity of the water. Draw $ab$ parallel to $ar$, and $de$ parallel to a tangent to the vane at the point $P$.

Then $Pe$ is the radius passing through $P$. $ar - \varphi r = v \tan \theta$, and $\varphi r = ar - v \tan \theta$, where $\theta$ is the angle made by the vane with the radius. Now if $\varphi$ is to be constant, $\varphi = a - \frac{v}{r} \tan \theta$ = constant.

$$\tan \theta = \left(\frac{a - \varphi}{v}\right) r = cr = \frac{rd\delta}{dr}; \quad \delta = \left(\frac{a - \varphi}{v}\right) r.$$

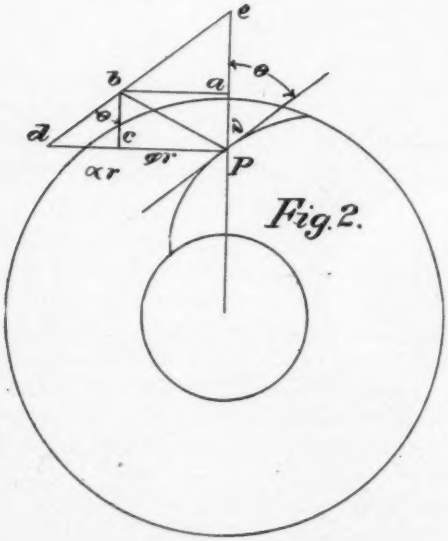



Fig. 2.

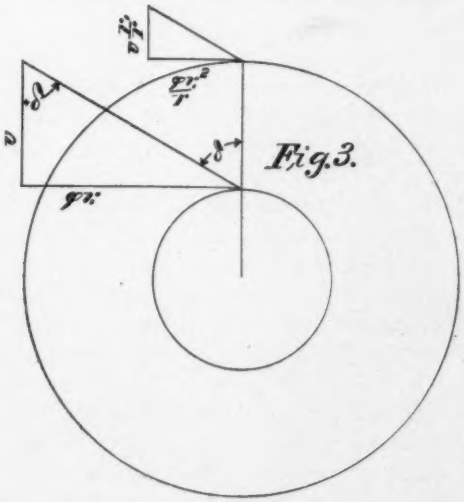
This equation is that of an archimedean spiral.

The continuity of the liquid does not require that this form of vane should be used, but as it gives the water constant angular velocity, there can be no fluid friction between the various layers, and consequently no eddy making due to this cause. On the other hand, there is shock at the inner end of the vanes, but this might be avoided by using guide vanes in the inlet nozzle. The amount of curvature, however, offers a ready means of regulating or varying the speed at which we wish to drive the wheel. It will be noted that the more nearly the relative path of the

water is normal to the vane the less the relative velocity of sliding along the vane, and consequently the less the friction.

All the formulæ in the above apply to radial vanes, in which case $\alpha = \varphi$.

There is still one point to be determined concerning this form of pump, *i.e.*, the form of the discharge conduit. When this is made so as to utilize as much as possible the tangential velocity of the water, then the radius at the discharge nozzle $r_2 = r_1\sqrt{1 + \frac{\varphi^2 r_1^2}{v^2}}$, and since the velocities of the water tangential and radial vary alike, or $v = \frac{v_1 r_1}{r}$ and $\varphi r = \frac{\varphi r_1^2}{r}$, then, referring to Fig. 3, let $ac$ = tangential velocity at any point, and




$bc$ = radial velocity at the same point; let $\delta$ be the angle made by the path of the water with a radius; then

$$\tan \delta = \frac{ac}{bc} = \frac{\varphi r_1}{r} \times \frac{r}{v_1 r_1} = \frac{\varphi r_1}{v} = \text{constant.}$$

Taking polar co-ordinates, $\tan \delta = \cot \theta = \frac{rd\theta}{dr} = \frac{\varphi r_1}{v_1}$.

$$d\theta = \left(\frac{\varphi r_1}{v_1}\right)\frac{dr}{r},$$

$$\theta = \left(\frac{\varphi r_1}{v_1}\right)\log_e r_1 = c \log_e r_1.$$

This is the equation of a logarithmic spiral, and the outer boundary of the discharge conduit should have this form, and the tangent of the angle it makes with the radius should be

$$\tan \delta = \frac{\varphi r_1}{v_1} = \text{constant quantity.}$$

Other forms of vanes and casings will be considered in a future paper.

## XX.

# SERVE'S RIBBED BOILER TUBE.

By Passed Assistant Engineer G. S. Willits, U. S. Navy.

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In the type of boiler generally adopted for use in large sea-going vessels no radical change can be made, it is thought, that will add to its efficiency as a steam generator while conforming to the restrictions put upon it by the service for which the vessels are intended. The size and number of the boilers are limited by the space available in the hold, and increased quantities of steam and pressures, called for by modern engines, are furnished by supplying the fires with forced draft. This so-called improvement has—under existing circumstances—reached its limit, and the marine engineer, searching for development, can only change the construction of some parts of the boiler or abandon the type.

Within the last three or four years the tube which gives title to this article has been attracting increasing and very favorable attention from certain classes of steam users on both sides of the Atlantic. It has been put into and thoroughly tested in various kinds of fire-tube boilers, and the engineer may look upon it as a decided and important improvement—one that has passed the experimental stage.

For those who may not have seen the tube itself, or a cut of it, a satisfactory description can be briefly and easily given: The tube is cylindrical, and from its inner or fire surface a number of equidistant radial ribs, parallel to the axis, converge toward the center; the radial length of the ribs is about one-fifth of the external diameter of the tube; the number of ribs is eight for tubes of and above 2¾ inches external diameter, and seven for those under that size; the mean thickness of the ribs is about the same as that of the tube itself, tapering toward the center;

the ribs stop about 3 inches short of the end of the tube to allow for expanding; both steel and brass tubes are made.

The steel plates or strips are rolled of the width or widths required, the last passage between the rolls bringing the plate out with the ribs all complete. After trimming the edges the plate is bent to an approximate shape and then lap-welded on a fluted mandrel. Brass tubes are solid drawn.

When these tubes have their ends thickened to receive a thread and the ribs are allowed to extend clear to the tube ends, they form excellent stays, being much stronger longitudinally than plain tubes of the same thickness. That they are better able also to resist a pressure tending to cause collapse may possibly be conceded. When a tube collapses, deformation does not take place, as a rule, at or near either tube sheet, because these parts are held to their true shape by the circumference of the hole in the tube sheet; therefore the ends of the ribs of a Serve tube are held in their place by the tube ends, close to which they approach, and each rib thus assumes, in a measure, the properties of a beam supported at its ends and uniformly loaded. If this reasoning is correct, a ribbed tube of diminished thickness could be used in place of a plain one for the same pressure. Experiments in this direction would be interesting.

A six-day comparative test of the efficiency of the ribbed and plain tubes was made at Elizabethport, N. J., last spring in the presence of certain naval and other engineers, complete logs being carefully kept. The data collected is given in the accompanying table with the deductions therefrom. A careful inspection of the table will disclose some surprising results. Attention is called to the low temperature of the escaping gases when the Serve tube was used.

To account for the superior efficiency of this tube, the theory is advanced that the ribs break up the column of gases and extract the heat from all parts of it, whereas, in the plain tube, only the exterior of this column gives off its heat. If the column of gases is not broken up, the ribs, at any rate, reach in toward the hottest part of it, and becoming highly heated, transmit the heat

COMPARATIVE BOILER TRIALS WITH PLAIN AND WITH SERVE'S PATENT RIBBED STEEL BOILER TUBES, COVERING TWELVE DA
HOURS. CONDUCTED BY H.

| | Date of trials, April, 1891. | Draft in inches of water. | Average temperature of chimney gases per pyrometer. | Average temperature of feed water. | Pounds of feed-water evaporated during trial. | Pounds of coal fired during trial. | Pounds of ashes and unconsumed coal on grate and in ash-pit at end of trial. | Pounds of picked out unconsumed coal in this. | Pounds of ashes in ash-pit and on grate at end of trial. |
|---|---|---|---|---|---|---|---|---|---|
| Plain Tubes. | 6 | ½ | 679.05 | 191.03 | 4,230 | 855 | 194 | ..... | ..... |
| | 7 | ½ | 685.15 | 189.54 | 4,475 | 861 | 210½ | 119 | 91½ |
| | 8 | ½ | 800.6 | 186.72 | 8,325 | 1,388 | 239½ | 97½ | 142½ |
| | 9 | ½ | 856.4 | 186.60 | 8,380 | 1,400 | 259½ | 107½ | 152½ |
| | 10 | ½ + 1/32 | 990.0 | 187.30 | 8,460 | 1,777 | 298 | 143 | 155 |
| | 11 | ⅝ | 922.73 | 186.75 | 8,190 | 1,780 | 369½ | 173 | 196½ |
| Ribbed Tubes. | 15 | ½ less 1/32 | 395.6 | 198.11 | 5,025 | 745½ | 266½ | 183 | 83½ |
| | 16 | ½ | 492.45 | 194.27 | 5,250 | 655½ | 164 | 75½ | 88½ |
| | 17 | ½ | 468.78 | 186.3 | 10,830 | 1,487 | 325½ | 141½ | 184½ |
| | 18 | ½ | 438.22 | 191.77 | 11,050 | 1,394½ | 299½ | 144 | 155½ |
| | 20 | ⅝ | 502.0 | 179.1 | 12,180 | 1,808½ | 375½ | 172½ | 202½ |
| | 21 | ⅝ | 519.68 | 181.0 | 12,200 | 1,800 | 370 | 123½ | 226½ |
| | 23 | 1 1/16 | 742.73 | 183.21 | 13,935 | 1,879½ | 379 | 114½ | 224½ |
| | 25 | 1 1/16 | 678.33 | 170.0 | 14,000 | 2,144½ | 341 | 160½ | 182 |

THE RESULT CONDENSED

| Pounds of coal consumed per 100 pounds of water evaporated. | | | Economy of coal. | Increase in steam generated. | Increase by Serve's tubes over plain tubes. | Total evaporation. | Evaporation per pound of combustible. | Extreme evaporation in 8 hours. | Durat |
|---|---|---|---|---|---|---|---|---|---|
| Draft. | Plain tubes. | Ribbed tubes. | | | | | | | |
| | | | *Per cent.* | *Per cent.* | | | *Per cent.* | | |
| Natural draft ½ | 19.72 | 13.65 | 30.83 | 18.03 | Natural draft ½ | 18.03 | 57.54 | 8,460 lbs. for plain tubes. | Two 8-hour tests wi |
| Forced draft ½ | 16.7 | 13.21 | 21.08 | 30.97 | Forced draft ½ | 30.97 | 32.68 | 14,000 lbs. for ribbed tubes. | Two 8-hour tests wi |
| Forced draft ⅝ | 21.37 | 14.8 | 30.74 | 46.46 | Forced draft ⅝ | 46.46 | 46.84 | Increase 65.5 per cent. | One 8-hour test with<br>One 8-hour test with |

Each day's trial lasted 8 hours; 32.35 lbs. of wood and 150 lbs. of coal were used to kindle the fire and to raise steam to 70 lbs. pressure, after which the trial immediately
points at the end of the trial. In trial of 25th alone steam was kept at 100 lbs. pressure in order to obtain the high draft and dry steam. The water level was also about 3 in
rials was 3 lbs. The last two days' trial was with ribbed tubes alone, to determine the limit of efficiency in comparison with plain tubes which was reached with the latter at
The quantities and results are taken as they are marked on the daily logs, without any calculated reductions. The boiler was a usual upright tubular of 42 inches diameter

G TWELVE DAYS, SIX DAYS WITH PLAIN AND SIX DAYS WITH RIBBED TUBES, EACH DAY'S TRIAL I
UCTED BY H. B. ROELKER, M. E.

| Pounds of ashes in ash-pit and on grate at end of trial. | Pounds of ashes per 100 pounds of coal. | Pounds of combustible consumed during trial. | Pounds of water evaporated per pound of coal fired. | Pounds of water evaporated per pound of combustible. | Pounds of combustible consumed per square foot of grate in trial. | Weather. | Barom |
|---|---|---|---|---|---|---|---|
| ..... | ..... | 676 | 4.95 | 6.28 | 12.05 | Clear day ; very light breeze. | 29.9 |
| 91½ | 9.05 | 665 | 5.50 | 6.73 | 11.87 | Clear day ; still. | 29.9 |
| 142½ | 9.27 | 1,162½ | 5.997 | 7.16 | 20.17 | Clear day. | 30.2 |
| 152½ | 9.84 | 1,154½ | 5.985 | 7.26 | 20.61 | Clear ; very light breeze. | 30.4 |
| 155 | 7.99 | 1,493 | 4.76 | 5.66 | 26.67 | Dull ; no rain. | 30.5 |
| 196½ | 10.15 | 1,425 | 4.60 | 5.74 | 25.44 | Rainy day ; coal wet ; ashes damp. | 30.1 |
| 83½ | 9.3 | 493½ | 6.74 | 10.10 | 8.80 | Fair. | 29.7 |
| 88½ | 11.01 | 505½ | 8.01 | 10.38 | 9.02 | Fair. | 29.9 |
| 184½ | 11.25 | 1,176 | 7.28 | 9.21 | 21.0 | Clear. | 30.0 |
| 155½ | 10.08 | 1,108½ | 7.92 | 9.96 | 19.80 | Clear. | 29.8 |
| 202½ | 10.34 | 1,447½ | 6.73 | 8.41 | 25 84 | Clear. | 30.1 |
| 226½ | 11.60 | 1,464 | 6.77 | 8.33 | 26.14 | Cloudy. | 30.3 |
| 224½ | 11.037 | 1,554½ | 7.41 | 8.32 | 27.57 | Clear. | 29.5 |
| 182 | 7.93 | 1,815½ | 6.32 | 7.71 | 32.41 | Fair. | 29.5 |

LT CONDENSED.

| Duration of Trials. | Pounds of feed water evaporated in both tests. | | Average temperature of feed water. | | Pounds of coal used in both tests. | |
|---|---|---|---|---|---|---|
| | Plain tubes. | Ribbed tubes. | Plain tubes. | Ribbed tubes. | Plain tubes. | Ribbed tubes. |
| Two 8-hour tests with natural draft, ⅛ in. water. | 8,705 | 10,275 | 190.28 | 196.19 | 1,716 | 1,401 |
| Two 8-hour tests with forced draft, ¼ in. water. | 16,705 | 21,880 | 186.66 | 189.03 | 2,788 | 2,881.5 |
| Two 8-hour tests with forced draft, ½ in. water. | 16,650 | 24,380 | 187.02 | 180.05 | 3,557 | 3,608.75 |
| One 8-hour test with forced draft, 1 7/16 in. water. | ..... | 13,935 | ..... | 183.21 | ..... | 1,879.5 |
| One 8-hour test with forced draft, 1 7/16 in. water. | ..... | 14,000 | ..... | 170.0 | ..... | 2,144.5 |

e trial immediately commenced. Steam was kept at 70 lbs. pressure and the water level at 7 inches in the glass, as nearly as possible, and b
was also about 3 inches in the glass. The soot scraped out of the plain tubes after the 6 days was 2½ lbs. That scraped from the Serve tubes
d with the latter at ½ inch draft pressure.
42 inches diameter with 36 inch furnace, 24 inches high, and with 63 tubes 2½ in. O. D. × 6 feet long ; 7 square feet grate surface ; 287 square

ILER TRIALS WITH PLAIN AND WITH SERVE'S PATENT RIBBED STEEL BOILER TUBES, COVERING TWELVE DAYS, SIX DAYS WITH
HOURS. CONDUCTED BY H. B. ROELKER, M. E.

| Draft in inches of water. | Average temperature of chimney gases per pyrometer. | Average temperature of feed water. | Pounds of feed-water evaporated during trial. | Pounds of coal fired during trial. | Pounds of ashes and unconsumed coal on grate and in ash-pit at end of trial. | Pounds of picked out unconsumed coal in this. | Pounds of ashes in ash-pit and on grate at end of trial. | Pounds of ashes per 100 pounds of coal. | Pounds o bustible sumed trial |
|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | 679.05 | 191.03 | 4,230 | 855 | 194 | ..... | ..... | ..... | 67 |
| $\frac{1}{4}$ | 685.15 | 189.54 | 4,475 | 861 | 210½ | 119 | 91½ | 9.05 | 66 |
| $\frac{1}{4}$ | 800.6 | 186.72 | 8,325 | 1,388 | 239½ | 97½ | 142½ | 9.27 | 1,16 |
| $\frac{1}{4}$ | 856.4 | 186.60 | 8,380 | 1,400 | 259½ | 107½ | 152½ | 9.84 | 1,15 |
| $\frac{1}{16} + \frac{1}{32}$ | 990.0 | 187.30 | 8,460 | 1,777 | 298 | 143 | 155 | 7.99 | 1,49 |
| $\frac{1}{8}$ | 922.73 | 186.75 | 8,190 | 1,780 | 369½ | 173 | 196½ | 10.15 | 1,49 |
| $\frac{1}{4}$ less $\frac{1}{32}$ | 393.6 | 198.11 | 5,025 | 745½ | 266½ | 183 | 83½ | 9.3 | 49 |
| $\frac{1}{4}$ | 492.45 | 194.27 | 5,250 | 655½ | 164 | 75½ | 88½ | 11.01 | 50 |
| $\frac{1}{4}$ | 468.78 | 186.3 | 10,830 | 1,487 | 325½ | 141½ | 184½ | 11.25 | 1,17 |
| $\frac{1}{4}$ | 438.22 | 191.77 | 11,050 | 1,394½ | 299½ | 144 | 155½ | 10.08 | 1,10 |
| $\frac{3}{8}$ | 502.0 | 179.1 | 12,180 | 1,808½ | 375½ | 179½ | 202½ | 10.34 | 1,44 |
| $\frac{3}{8}$ | 519.68 | 181.0 | 12,200 | 1,800 | 330 | 123½ | 226½ | 11.60 | 1,46 |
| $1\frac{1}{16}$ | 742.73 | 183.21 | 13,935 | 1,879½ | 339 | 114½ | 224½ | 11.037 | 1,55 |
| $1\frac{1}{16}$ | 678.33 | 170.0 | 14,000 | 2,144½ | 341 | 160½ | 182 | 7.93 | 1,81 |

THE RESULT CONDENSED.

| consumed per 100 pounds of ater evaporated. | | Economy of coal. | Increase in steam generated. | Increase by Serve's tubes over plain tubes. | Total evaporation. | Evaporation per pound of combustible. | Extreme evaporation in 8 hours. | Duration of Trials. |
|---|---|---|---|---|---|---|---|---|
| Plain tubes. | Ribbed tubes. | | | | | | | |
| | | *Per cent.* | *Per cent.* | | | *Per cent.* | | |
| 19.72 | 13.65 | 30.83 | 18.03 | Natural draft $\frac{1}{4}$ | 18.03 | 57.54 | 8,460 lbs. for plain tubes. | Two 8-hour tests with natural draft, $\frac{1}{4}$ in. water. |
| 16.7 | 13.21 | 21.08 | 30.97 | Forced draft $\frac{1}{4}$ | 30.97 | 32.68 | 14,000 lbs. for ribbed tubes. | Two 8-hour tests with forced draft, $\frac{1}{4}$ in. water. |
| 21.37 | 14.8 | 30.74 | 46.46 | Forced draft $\frac{3}{8}$ | 46.46 | 46.84 | Increase 65.5 per cent. | Two 8-hour tests with forced draft, $\frac{3}{8}$ in. water. |
| | | | | | | | | One 8-hour test with forced draft, $1\frac{1}{16}$ in. water. |
| | | | | | | | | One 8-hour test with forced draft, $1\frac{1}{16}$ in water. |

asted 8 hours; 32.35 lbs. of wood and 150 lbs. of coal were used to kindle the fire and to raise steam to 70 lbs. pressure, after which the trial immediately commenced. Steam was kept
trial. In trial of 25th alone steam was kept at 100 lbs. pressure in order to obtain the high draft and dry steam. The water level was also about 3 inches in the glass. The soot
st two days' trial was with ribbed tubes alone, to determine the limit of efficiency in comparison with plain tubes which was reached with the latter at $\frac{1}{4}$ inch draft pressure.
l results are taken as they are marked on the daily logs, without any calculated reductions. The boiler was a usual upright tubular of 42 inches diameter with 36 inch furnace, 24 inche

K DAYS WITH PLAIN AND SIX DAYS WITH RIBBED TUBES, EACH DAY'S TRIAL LASTING EIGHT
KER, M. E.

| of ashes pounds oal. | Pounds of combustible consumed during trial. | Pounds of water evaporated per pound of coal fired. | Pounds of water evaporated per pound of combustible. | Pounds of combustible consumed per square foot of grate in trial. | Weather. | Barometer. | Outside temperature. |
|---|---|---|---|---|---|---|---|
| .... | 676 | 4.95 | 6.28 | 12.05 | Clear day; very light breeze. | 29.91 | 42° |
| 05 | 665 | 5.20 | 6.73 | 11.87 | Clear day; still. | 29 98 | 42° |
| 27 | 1,162¼ | 5.997 | 7.16 | 20.17 | Clear day. | 30.23 | 48° |
| 84 | 1,154¼ | 5.985 | 7.26 | 20.61 | Clear; very light breeze. | 30 46 | 51° |
| 99 | 1,493 | 4.76 | 5.66 | 26.67 | Dull; no rain. | 30.53 | 44° |
| 15 | 1,425 | 4.60 | 5.74 | 25.44 | Rainy day; coal wet; ashes damp. | 30.17 | 53° |
| 3 | 493½ | 6.74 | 10.10 | 8.80 | Fair. | 29.78 | 73° |
| 01 | 505½ | 8.01 | 10.38 | 9.02 | Fair. | 29.98 | 60° |
| 25 | 1,176 | 7.28 | 9.21 | 21.0 | Clear. | 30.09 | 62° |
| 08 | 1,108¾ | 7.92 | 9.96 | 19.80 | Clear. | 29.83 | 59° |
| 34 | 1,447½ | 6.73 | 8.41 | 25 84 | Clear. | 30.15 | 63° |
| 60 | 1,464 | 6.77 | 8.33 | 26.14 | Cloudy. | 30.36 | 53° |
| 037 | 1,554½ | 7.41 | 8.32 | 27.57 | Clear. | 29.56 | 70° |
| 93 | 1,815¾ | 6.52 | 7.71 | 32.41 | Fair. | 29.58 | 52° |

| RIALS. | Pounds of feed water evaporated in both tests. | | Average temperature of feed water. | | Pounds of coal used in both tests. | | Pounds of water evaporated per pound of coal. | |
|---|---|---|---|---|---|---|---|---|
| | Plain tubes. | Ribbed tubes. | Plain tubes. | Ribbed tubes. | Plain tubes. | Ribbed tubes. | Plain tubes. | Ribbed tubes. |
| draft, ⅛ in. water. | 8,705 | 10,275 | 190.28 | 196.19 | 1,716 | 1,401 | 5.08 | 7.35 |
| draft, ¼ in. water. | 16,705 | 21,880 | 186.66 | 189.03 | 2,788 | 2,881.5 | 5.98 | 7.60 |
| draft, ½ in. water. | 16,650 | 24,380 | 187.02 | 180.05 | 3,557 | 3,608.75 | 4.68 | 6.75 |
| aft, 1⅛ in. water. | ..... | 13,935 | ..... | 183.21 | ..... | 1,879.5 | ..... | 7.41 |
| raft, 1⅛ in water. | ..... | 14,000 | ..... | 170.0 | ..... | 2,144.5 | ..... | 6.52 |

d. Steam was kept at 70 lbs. pressure and the water level at 7 inches in the glass, as nearly as possible, and both were at these
glass. The soot scraped out of the plain tubes after the 6 days was 2½ lbs. That scraped from the Serve tubes after the 6 days'
t pressure.
ch furnace, 24 inches high, and with 63 tubes 2½ in. O. D. × 6 feet long; 7 square feet grate surface; 287 square feet fire surface.

thus acquired through their substance to the water surface of the tube.

It is claimed that less soot and cinder are deposited, considering the extra surface, etc., than in the plain tube. Be that as it may, the tubes are easily cleaned. The writer saw chalk marks in the angles between the ribs and the cylindrical surface removed by one passage through the tube of an ordinary tube brush.

Two objections may be urged against the ribbed tube, the increased cost and the extra weight. As to the greater cost—three to four times—the superior economy of this tube soon removes that disadvantage, and there remains only the extra weight to be considered. This weight is, roughly speaking, about fifty per cent. greater than with the same size of plain tube, but in a boiler having stay tubes this excess of weight will be considerably less than that figure. If the extra weight becomes the more important objection the required efficiency may be obtained with fewer tubes, or a smaller boiler, or, in a large battery, by removing one or more boilers.

A specially designed stopper is required to plug a leaky tube.

[In this connection attention is called to information on the Serve tube and comparative tests published in No. 4, Vol. II, of the JOURNAL, p. 569. The tests referred to show a much smaller gain, both in amount and in efficiency of evaporation. The boiler experimented on in the trials now reported is one which is very wasteful of fuel unless the tubes are of great length. Had the weight added by the ribbed tubes been used in extending the length of the plain tubes, the results would not have shown differences so enormous, though certainly still greater than those recorded of boilers with horizontal tubes.—ED.]

## XXI.

## WEBSTER'S CLINOMETER.

The instrument here described and illustrated is the invention of P. A. Engineer H. Webster, U. S. Navy, and is intended to replace the pendulum as a device for indicating the roll of a ship.

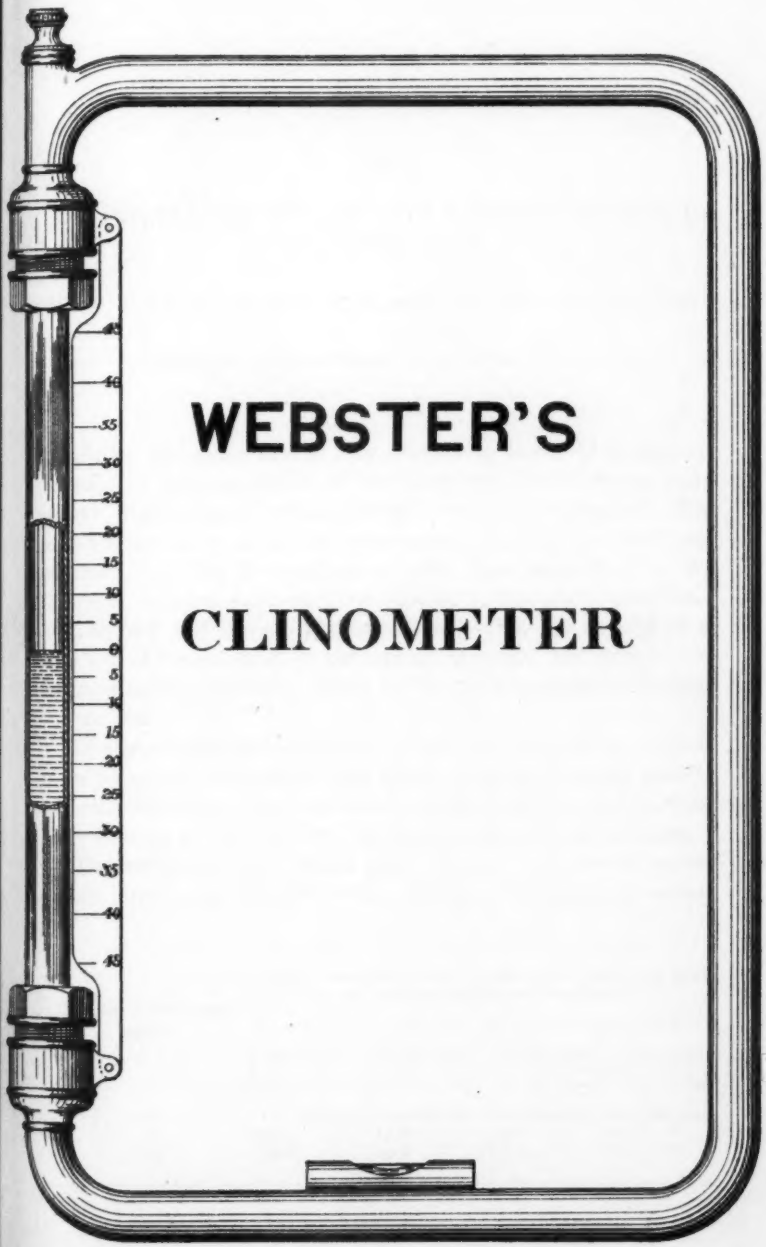
The details of the instrument are very simple, and its method of operation is evident by inspection. Three sides of the rectangular system are of metal tube, the fourth side being of glass.

Charging the system with a liquid to the zero point on the appended scale, the device is ready for operation. Inasmuch as liquids tend to maintain a level, it is manifest that if the system is moved out of the plane of the horizon, such movement will be indicated by the changed position of the surface of the fluid in the glass column. And if the appended scale is accurately marked to the proper radius, such deviations of the system will be shown in terms of arc.

Secured in an athwartship plane on a vessel afloat, any roll, however small or however slow, will be accurately marked by this instrument.

In practice this clinometer is found to operate without error from friction, "send" or "pumping."

The small spirit level on the lower part of the clinometer is for the purpose of testing the accuracy of the zero level of the fluid column.



# WEBSTER'S

# CLINOMETER

## XXII.

# ARTIFICIAL DRAFT AND ITS EFFECTS ON BOILER CONSTRUCTION.

By E. Lechner, Marine Architect, German Imperial Navy.

[Translated by Assistant Engineer Emil Theiss, U. S. Navy.]

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Under the above title a very interesting series of papers appeared in the June issues of the Zeitschrift des Vereines Deutscher Ingenieure, of which the following is an abstract. The conclusions arrived at are interesting, not only on their own account, but also in comparison with those arrived at by Mr. Yarrow, which they in part corroborate and in part contradict.

The first and foremost consideration which led and leads to the use of forced draft is the saving of space and weight. A second rather incidental object is the more complete utilization of the fuel.

Of experiments made during the last years to perfect combustion by means of artificial draft those made by Howden deserve special mention. They are based on the principle of furnishing just enough air to consume the fuel completely and no more.

The subjoined table makes apparent the losses due to excess of air introduced into the furnace, and is calculated for complete

| Temperature of escape gases. | Loss of heat in per cent. of fuel burnt due to heat in the escape gases with an excess of air of | | | |
|---|---|---|---|---|
| | 0 per cent. | 25 per cent. | 50 per cent. | 75 per cent. |
| 482 | 11.9 per cent. | 14.6 per cent. | 20.1 per cent. | 36.5 per cent. |
| 572 | 13.7 " | 17.0 " | 23.6 " | 43.3 " |
| 662 | 15.6 " | 19.6 " | 27.1 " | 50.1 " |

combustion and for a fuel consisting of 75 per cent. carbon; 3.5 per cent. free hydrogen; 13 per cent. of water chemically combined; 4 per cent. of hygroscopic water, and 4.5 per cent. ash.

Insufficient air supply, of course, leads to losses also, the carbon burning to carbonic oxide instead of carbonic acid. Howden, Hoadley and Spence have made the attempt, by using centrifugal fans, in a manner to apportion to the furnace a fixed quantity of air; at the same time they turned the waste heat to account by heating the air supplied for combustion. Howden calculated the gain to be made in this way at 50 per cent., but his own published accounts of trials, and those of Hoadley and Spence, do not show gains nearly as high. On thirty-one trials with natural draft Spence found an efficiency of the boiler of 68.7 per cent. Eighteen trials with artificial draft, without heating the air, gave an efficiency of 68.6 per cent. Seven trials with artificial draft and heated air gave 76.7 per cent. Increase of efficiency was, therefore, obtained by heating the air supplied for combustion, not by the artificial air supply. Regulating the air supply in exact accordance with the amount required at any time will become a possibility only when the chemical composition of the escape gases can at any time be found by reference to a simple apparatus, either automatically recording, or at any rate, readily manipulated. (An instrument of this kind is in existence and in practical use, but is too delicate and fragile for use on board ship. It comprises a pressure gauge and a pyrometer, and the per cent. of carbonic acid in the escape gases is registered in per cent. of the air pressure and temperature.)

Although little or nothing seems to have been gained so far by these attempts to regulate the air supply, yet a positive gain is effected by transferring the chimney work to the blowers. With natural draft this work absorbs from $\frac{1}{9}$ to $\frac{1}{4}$ of the heat utilized. In a ship instanced, fitted with centrifugal blowers, the steam consumed by them was less than $\frac{1}{44}$ of the total steam generated.

With artificial draft it is possible to utilize the heat of the escape gases for heating the air supply or the feed water. But as the temperatures fall lower, the heating surface required for ab-

sorbtion becomes correspondingly greater, so that a limit is soon reached beyond which it is impossible to go without sacrificing the main object of forced draft, decrease in weight. Advantages claimed for artificial draft on the score of economy of fuel can, therefore, scarcely be allowed to exist.

The reduction in weight due to the introduction of forced draft is undoubtedly great. While twenty years ago the fire-room weights aggregated about 275 lbs. per I.H.P., these weights have on modern torpedo boats been reduced to 40 lbs. Even making allowance for gain due to increase of pressure and better material and workmanship these weights would, without forced draft, be 3 or 4 times those given. The gain is, of course, proportional to the degree of forcing adopted. This it is customary to measure in pressure in inches of water in the fire room or the ash-pit—an uncertain guide, because it does not furnish a means of estimating accurately the coal consumed per unit of grate area.

A study of the results of trial trips shows that the air pressure carried seems to have no fixed relation with the amount of coal burnt per square foot per hour. If these discrepancies existed only between boilers of different type this difference might be adduced as a reason. But here is a German ship, on which, the crew remaining unchanged, a pressure of 2.36 in. had to be maintained on a six-hour trial to burn 27.4 lbs. of coal per square foot per hour and develop 1,798 I.H.P., while a few days later on a similar official six-hour trial, under similar conditions, with an air pressure only one-half as great, 26.1 lbs. were burnt per square foot and 1,733 I.H.P. developed. On another ship, during the six-hour trial for acceptance, 2.44 inches of pressure were maintained in one fire room, while in another one exactly similar 4.09 inches had to be maintained. These discrepancies are made apparent in the table on the following page.

All these ships have four double-ended boilers with three furnaces at each end. But for unimportant details the construction is identical, except that while the *Australia* and *Galatea* have one combustion chamber for all six furnaces of a boiler, the rest have three separate combustion chambers for each boiler,

opposite furnaces having one in common. Nevertheless the air pressure required for approximately equal performances varies from 1.02 to 2.01 inches. In view of the serious effects on a boiler of high air pressures carried an examination of the relations existing between pressure and fuel consumption is important.

| Name of ship. | Builder. | Mean air pressure in fire rooms. | Mean I.H.P. per sq. ft. of grate surface. |
|---|---|---|---|
| *Orlando* | Palmer. | 1.02 | 16.17 |
| *Undaunted* | ..... | 1.87 | 16.17 |
| *Australia* | Napier. | 1.77 | 17.75 |
| *Galatea* | ..... | 1.14 | 18.50 |
| *Narcissus* | Earle. | 1.02 | 16 17 |
| *Immortalite* | ..... | 2.01 | 17.93 |

The quantity of air theoretically necessary is proportional to the coal burnt. Assuming that for a given boiler with a grate of a certain construction the coal is carried at a uniform height, the spaces between the grate bars and between the fuel through which the air must pass will remain the same, and we should have $m = \sqrt{c(h_1 - h_2)}$, where $m$ is the quantity of air (or the quantity of coal per square foot per hour, to which the air supply is proportional), $h_1$ and $h_2$ are the pressures on the two sides of the grate, *i. e.*, in the fire room or the ash pit and in the furnace, and $c$ is a constant depending for its value on the air space through the grate and the layer of fuel, on the height of the barometer, the temperature of the air, the rise in temperature of the air in passing through the fuel, &c.

The air pressure required is therefore given by $h_1 - h_2 = \frac{m^2}{c}$; *i. e.*, the layer of fuel remaining of constant depth, the pressure required to burn a certain quantity of coal is proportional to the square of the quantity to be burnt per square foot per hour.

Experience has demonstrated, however, that to burn an increased quantity of coal per square foot per hour economically, it is necessary to increase the thickness of the layer of fuel.

With increase of this thickness goes a decrease of the space left for the air to pass through and a corresponding decrease of $c$. The air pressure, or rather the difference of pressure outside and in the furnace, must therefore increase more rapidly than in proportion to the square of the fuel burnt per square foot per hour. In general it may be assumed that the penetrability of a layer of fuel by the air is inversely proportional to its depth, and, leaving height of barometer and temperature of the air out of account, it follows that the air pressure difference must increase in proportion to the product of the square of the fuel burnt per square foot per hour and the depth of the fuel on the grate.

As the area of the passages traversed by the gases beyond the furnace remain unaltered, and the quantity of escape gases is proportional to the fuel burnt, we have for the pressure in the furnace $h_2 = \frac{m^2}{c_1}$, $c_1$ being a constant depending on the area of cross-section of the flue and tubes, the height of barometer, &c.

The correctness of these assumptions is evidenced by the following table, giving results obtained on a test of the locomotive boiler of a torpedo boat:

| 1. | 2. | 3. | 4. | 5. | 6. | | | 7. | 8. | 9. | 10. | 11. | 12. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Number of trial. | Grate surface, sq. feet. | Heating surface, sq. feet. | Ratio of heating to grate surface. | Boiler pressure above atmosphere, lbs. per sq. inch. | Air pressure in inches of water. | | | Coal burnt per hour per sq. foot of grate, lbs. | Feed-water evaporated per hour from a temperature of 86° F., lbs. | Water evaporated per lb. of coal, lbs. | Temperature of chimney gases, deg. F. | Air supplied per minute, cubic feet. | Mean thickness of fuel layer on grate, inches. |
| | | | | | Fire room $h_1 =$ | Furnace $h_2 =$ | Chimney $h_3 =$ | | | | | | |
| *A*, all the grate surface in use; artificial draft by centrifugal fan in closed fire room. | | | | | | | | | | | | | |
| 1 | 20.44 | 882.32 | 43.1 | 113.6 | 1.97 | 1.57 | .39 | 53.6 | 437.8 | 8.16 | 518 | 4,112 | 12.60 |
| 2 | ..... | ..... | ..... | ..... | 2.93 | 2.36 | .59 | 63.4 | 517.5 | 8.17 | 608 | 5,365 | 13.40 |
| 3 | ..... | ..... | ..... | ..... | 3.94 | 3.15 | .91 | 76.4 | 583.6 | 7.64 | 626 | 6,287 | 14.17 |
| 4 | ..... | ..... | ..... | ..... | 5.90 | 4.53 | 1.18 | 93.4 | 649.8 | 7.00 | 716 | 7,092 | 15.35 |
| *B*, half the grate surface in use. | | | | | | | | | | | | | |
| 5 | 10.22 | 882.32 | 86.2 | 113.6 | 2.95 | 2.17 | .39 | 66.7 | 528.6 | 7.93 | 554 | 5,221 | 9.84 |
| 6 | ..... | ..... | ..... | ..... | 5.90 | 3.94 | 1.00 | 101.4 | 710.0 | 7.00 | 608 | 6,831 | 11.82 |
| 7 | ..... | ..... | ..... | ..... | 6.80 | 4.53 | 1.18 | 113.0 | 770.2 | 6.80 | 698 | 7,092 | 13.78 |

REMARK.—The coal was capable of evaporating 9.06 pounds of water from and at 212° F. The refuse contained amounted to 5.7 per cent., of which 3.92 per cent. was clinker, as determined in an experimental boiler.

The trials, each of which extended over several hours, were made for the purpose of determining the capacity of the boiler under forced draft. During each trial the air pressure was kept constant in the closed fire room (at 1.97, 2.95, 3.94, 5.90 and 6.89 inches), and the trials were made with all and with one-half the grate surface in use, in order to determine the effect of the change of ratio $\frac{\text{heating surface}}{\text{grate surface}}$.

Although all the trials extended over six hours, with the same men and the same coal, yet it is evident that the values in columns 6 and 7 may need correction, for even a slight change in the size of the lumps of coal used and the slack contained would have an immediate effect on the relation obtaining between air pressure and coal consumed. These values were therefore plotted with weights of coal consumed per hour per square foot of grate as abscissæ and corresponding air pressures as ordinates, and smooth curves constructed. The subtraction of corresponding ordinates $h_1$ and $h_2$ gave a curve of pressure differences.

From the curves we get, for example, for abscissæ equal to 61.3 and 92.0, pressures in fire-room $=H_1$ and $h_1 = 6.06$ and 2.62; and in the furnace $=H_2$ and $h_2 = 4.57$ and 2.05.

Hence $\frac{H_2}{h_2} = \frac{4.57}{2.05} = 2.23$ and $\frac{H_1 - H_2}{h_1 - h_2} = \frac{1.49}{.57} = 2.62$.

The ratio of the squares of the corresponding abscissæ is, $\frac{\overline{92.0}^2}{\overline{61.3}^2} = \left(\frac{3}{2}\right)^2 = 2.25$.

For these coal consumptions column 12 of the table gives a thickness of fuel on the grate of 13.39 and 15.35 inches. In *B* group we get for these values by interpolation 9.06 and 10.63 inches. If the theoretical conclusions arrived at above are correct, then $\frac{H_1 - H_2}{h_1 - h_2} = \frac{\overline{92.0}^2}{\overline{61.3}^2} \times \frac{15.35}{13.39}$ and also $= \frac{\overline{92.0}^2}{\overline{61.3}^2} \times \frac{10.63}{9.06}$.

These values work out 2.58 and 2.64, which are close enough to the value 2.62 obtained from the trial data to establish the truth of the connection pronounced above to exist between air pressure and coal consumption per unit of grate surface.

The connection has been shown to exist, and consequently to hold, only for a certain type of boiler, for a certain quality of coal, and for similar firing, and we have to show now the influence exercised on this connection by

1. The number, size, and grouping of the boilers of a ship;
2. The differences in the details of construction of different boiler types;
3. The diverse qualities of the coals used ordinarily;
4. Differences in the firing.

To determine the influences on the relations existing between air pressure and coal consumption, of number, size, and grouping of boilers, it is necessary to compare the performance of different ships that widely differ in these respects.

The following table is compiled for this object: The area of the grate gives an idea of the size of boilers, while the number of fire rooms gives an idea of the grouping:

| Vessel. | Number of boilers. | Total grate surface. | Number of boiler rooms. | Grate surface in each boiler room. | Trial trip results. | |
|---|---|---|---|---|---|---|
| | | | | | Mean air pressure in inches of water. | Coal consumed per square foot per hour. |
| I | 1 | 8.07 | 1 | 8.07 | 3.15 | 85.85 |
| II | 1 | 20.44 | 1 | 80.44 | 3.15 | 66.43 |
| III | 4 | 162.48 | 2 | 21.24 | 3.27 | 58.05 |
| IV | 4 | 645.60 | 2 | 322.80 | 3.11 | 33.73 |

Of the above vessels, I is a steam launch with a small locomotive boiler. No. II is a torpedo boat with a somewhat larger locomotive boiler; No. III is a dispatch boat with four large locomotive boilers with two grates each, placed in two separate compartments; No. IV is a naval vessel with four large six-furnace, double-ended boilers in two separate compartments. The above table shows conclusively that the greater the size and number of boilers, or the greater the grate surface crowded into one compartment and supplied by the same blowers, the less

becomes the coal consumed per hour per square foot of grate for the same air pressure; or the greater must be the air pressure to burn a certain amount of coal per square foot per hour. If we therefore wish to avoid high air pressures and the dangers incurred by their use, it is essential to create as many separate fire rooms as possible. This conclusion appears surprising, but it is what ought to be expected, as will become apparent later on.

As regards differences in construction of boilers and their effects on the relation in question, the only differences to be taken account of are the size of the ash pit, the free area through the grate, over the bridgewall, through the tubes, the connections and the chimney, in proportion to the grate surface. With natural draft the following ratios were accepted in practice:

1. $\frac{\text{Opening into ash-pit}}{\text{grate surface}} = .11 \text{ to } .30.$

2. $\frac{\text{Area over bridgewall}}{\text{grate surface}} = .14 \text{ to } .35.$

3. $\frac{\text{Smallest area through tubes}}{\text{grate surface}} = .14 \text{ to } .20.$

4. $\frac{\text{Clear area of smoke pipe}}{\text{grate furface}} = .10 \text{ to } .20.$

No ratio is given for the free area through the grate, because the diversity of practice is too great.

Calculating the same quantity for some locomotive boilers worked with forced draft we get the following table:

| No. | By whom built. | Grate surface. | Ratio of heating to grate surface. | Ratios of passages traversed by air and gases to grate surface. | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | Ash-pit. | Bridgewall. | Tubes without ferrules. | Tubes with ferrules. | Smoke pipe. |
| 1 | Yarrow..... | 27.00 | 42.6 | .07 | .28 | .15 | .10 | .16 |
| 2 | Thornycroft..... | 30.13 | 37.7 | .16 | .22 | .16 | .12 | .13 |
| 3 | Germania Co..... | 24.75 | 43.9 | .20 (about) | .30 (about) | .15 | .10 | .18 (about) |

A number of larger cylindrical marine boilers constructed for forced draft, gave ratios differing little from those given. Owing to the cylindrical form of the furnaces the area over the bridge-wall is somewhat smaller than in the locomotive boilers, the ratio being from .15 to .19.

We see from what precedes that the introduction of forced draft left the construction of boilers practically unchanged as regards the flue areas. The only ratio that we find seriously changed is the ratio of area through tubes to grate surface, when the tubes are ferruled. The increased resistance met with by the escape gases demands higher air pressure to burn a certain quantity of coal per square foot per hour, for with two boilers with different ratio $\frac{\text{area through tubes}}{\text{grate surface}}$, but otherwise identical and burning the same amount of coal, the air pressure to be carried must be inversely proportional to the square of that ratio. It is for that reason that ferruling tubes is to be avoided, for though ferrules may stop the leakage of tubes temporarily, the remedy aggravates the evil by necessitating higher air pressure and augmenting the cause producing leakage.

With brass tubes, however, steel ferrules have been found a necessity to prevent leakage, no matter whether the tubes are simply expanded into the tube sheets or screwed in. The locomotive boilers built by Schichau had formerly iron and have now steel tubes, without ferrules. The practice of stopping leakage of tubes by driving in ferrules had been followed, but it was soon discovered that instead of improving they made matters worse, and they were removed. It may, therefore, be stated as a fact based on experience, that ferrules are to be avoided, and the designer must so choose his material and proportion his parts as to make their use superfluous, and keep the area through the tubes as large as possible.

What has been said concerning the area through the tubes applies as well to the area over the bridgewall and through the connections and the smoke pipe. The latter are to be made of ample size for forced-draft boilers, and with cylindrical furnaces the area over the bridgewall may well be increased by moving it back into the combustion chamber and lowering the back of

the grate. This latter expedient, however, involves a new danger which will be dwelt on later.

Of no less importance than the above conditions is the quality and size of the coal used. A somewhat bituminous, slightly caking coal of good heating power, with few incombustible constituents, of a size of lump from 1½ to 4 inches, is the best for forced draft. Hard coal is less desirable, and coal that crumbles is entirely unsuited, because it disintegrates in the furnace and the strong draft carries it into the tubes, which it soon chokes, requiring a constant increase of air pressure to keep up the rate of combustion. The less the tendency of the coal to cake the more care must be taken to avoid slack, as it will choke up the tubes; but caking coal, too, must be fired with discretion, because if any slack is carried up into the tubes it sticks there, cakes and swells, and must in the end be removed by drilling. It stands to reason that freedom of the coal from ash and clinker is of much greater importance when using forced than when using natural draft, for more coal is burnt per square foot of grate, ash and clinker accumulate more rapidly, and the fires must either be cleaned so frequently as to sacrifice the benefits derived from forcing the draft, or the air pressure must be increased as the grate becomes choked.

Ash and clinker are also at the bottom of the formation of what the Germans call "swallows' nests." These are accumulations of clinker, particles of half-burnt coal and ash, with perhaps a quantity of salt from the water leaking through, that form on the back tube-sheets. They take the form of rings, and gradually grow over and into the tube openings, which they eventually almost or entirely close, necessitating increased air pressure.

On the subject of the use of petroleum as a substitute for coal, the author holds, that though experiments with it on board ship have given good results, the sudden turning on and stopping of the jets works injury to the boiler. He considers its use on naval vessels out of the question, chiefly because of the danger involved in having large quantities of it on boaad during an engagement.

Coke the author dismisses as a fuel, because of its bulk and

its short flame of intense local heating power. The hydro-carbon gases of coal fuel fill a useful office in saving the boiler from too sudden changes of temperature when the furnace door is opened; for with the inrush of air they burn with a long flame and check the effects of the stream of cold air.

In what has been said frequent allusion has been made to the influence of the working of the fires on the air pressure to be carried, and, consequently, on the endurance of the boiler. The importance of this factor in the success or the failure of a boiler to do its work well cannot be overestimated. For a boiler to do its best with the minimum of effort the different operations of firing must be executed with certainty and despatch, and those conditions avoided which lead to sudden changes in the state of the fires or the rate of steam generation, such as keeping furnace doors open unduly long, irregularity in the air pressure, irregular feed and steam consumption.

The thickness of the fuel layer is to be regulated according to the rate of forcing adopted, and must be uniform all over the grate. Column 12, of the table on page 353, is a guide in a measure, but in every case it is necessary to take account of the quality of the coal and the amount of slack contained. The area of grate is another factor to be considered, for the greater this area in a boiler or in a compartment the heavier must be the fuel layer for a fixed rate of combustion, and the more care must be taken to distribute coal and slack evenly over the grates to prevent the violent rush of cold air through thinly covered spots. This is more readily prevented if in the original design care has been taken to make the grates of a shape and size as to be easily manageable and to distribute the air in numerous fine streams. With cylindrical furnaces heavy fires are difficult to attend properly through fire-doors necessarily low down, particularly if the coal burns, as it should do with forced draft, with a long flame. The area of such grates should not, therefore, exceed 21.5 square feet, nor the length 5.9 to 6.6 feet. With locomotive boilers the fire-doors are better located and the construction more favorable. With forced draft, however, the grate area should not exceed 37.7 to 43 square feet, because in this case as much as 3,000 to

3,500 lbs. of coal must be fired and worked per hour, which necessitates keeping the fire-doors open so much that the fire has scarcely time during the intervals to get well started. With large locomotive boilers it is therefore to be recommended to divide the grate and carry the division to the top of the chimney.

For the purpose of obtaining a finer distribution of the air supplied for combustion preference is to be given to grate bars of slender cross-section. The extreme position is held by the Ferrando grate, which is composed of bars $\frac{3}{8}$ inch thick, with spaces about $\frac{1}{8}$ inch wide. It is claimed that with such grates the fire can be carried much lighter, but it would seem as though they would become clogged with ash and clinker very quickly. Practice and theory unite in demanding that the air spaces shall be as large as possible, and for torpedo boats a triple bar, each single bar abont $\frac{3}{8}$ inch wide at top, tapering to about $\frac{3}{16}$ at bottom, the spaces tapering from $\frac{3}{8}$ at top to $\frac{9}{16}$ at bottom, and the depth about $3\frac{1}{8}$ inches, was speedily adopted.

For large cylindrical boilers the greater cross sections of grate bar formerly universal are still in use for forced draft, and the disadvantage is, that the bars must either be spaced far apart, allowing the coal to drop through, or else the air space through the grate must be decreased. Much coal dropping through not only throws extra work on the firemen, but leads to the overheating, warping and burning of the grate. The slender section recommended for torpedo-boat boilers is not suitable for the ordinary cylindrical type, as it is not expedient to carry water in the ash-pit in the latter case, a practice in vogue in torpedo-boat boilers. A section of about $3\frac{3}{8} \times \frac{3}{4} \times \frac{5}{16}$ inches would be more nearly the required size.

Speaking of the advantages claimed for the different systems of forced draft, the author describes an arrangement in use on the *City of Paris* before her breakdown, by which the gases from each furnace were kept separate by diaphragms until they reached the base of the smoke pipe, where each separate duct was provided with a damper, which was to be closed whenever the fire door was opened, and thereby prevent the inrush of cold air. To be easily accessible and manageable, these dampers

should be brought down into the front connections, just above the tubes. Furthermore, in speaking of the dangers from leakage of tubes, leading to the sudden formation of steam in the furnaces and combustion chambers, and finding, in company with the furnace gases, exit into the fire-room or up through the ducts and ventilators, according as closed fire-room or closed ash-pit draft is used, the author states that he has on three occasions seen the steam pressure drop 45 to 60 pounds in the space of a few seconds.

Having begun with a mathematical investigation of the advantages of forced draft, the author now proceeds to deal with the disadvantages that have had to be taken into the bargain, and that have so far been only partially overcome.

These disadvantages are treated under the heads of—

1. Leakage of tubes.
2. Splitting of tubes.
3. Flattening of tubes.
4. Bulging of tube sheets.
5. Breakage of stay bolts.
6. Leakage of rivet seams.
7. Melting of grate bars; wearing down of bridgewalls.
8. Overheating connections and smoke pipes; flames issuing from smoke pipes.
9. Flames issuing from the furnace mouths.

The last named evil, which in reality is only a result of excessive leakage of tubes, is confined chiefly to boilers of the locomotive type, because with other types the back tube sheet is so located, as a rule, that the spray of water issuing from the tube ends cannot fall directly upon the fire, but falls into the bottom of the combustion chamber instead, and finds its way out through the ash-pit. Single combustion chamber double-ended boilers, however, are liable to the same accident, and accordingly we hear of such catastrophes as that which occured on the *Barracouta*. In this case the hot gases driven into the fire-room by the steam pressure generated were fired on coming in contact with the air, and the whole fire-room was for about a minute ablaze with flame.

When forced draft first came into extensive use, it was considered of first importance, in view of the increased temperatures and the greater quantity of heat to be transmitted, to provide some metal for boiler material of high thermal conductivity. At the same time the higher steam pressures carried seemed to demand securer bracing. Accordingly, the different countries adopted generally and well nigh simultaneously copper for fire boxes, brass for tubes. Heavy iron braces were spaced among the tubes to brace the tube sheets. This construction was far from satisfactory, however, and repeated serious leakages of tubes made it evident that iron braces among brass tubes were inexpedient; for not only is the coefficient of expansion of iron only about 50 per cent. of that of brass, but while the tubes are exposed to the heat of the furnace gases, the braces are at the temperature of the water in the boiler. It happened not infrequently that, after a period of hard driving, tubes were found to have moved in the tube sheets from .02 to .04 inch in spite of expanding, beading and ferruling. Iron braces were therefore discarded, brass stay tubes of heavier gauge than the ordinary tubes were introduced, and all tubes were screwed into the back tube sheet wherever practicable. There was improvement, still the tubes leaked on occasions, and the leakage was confined mainly to the stay tubes and the ordinary tubes immediately adjoining.

An explanation is found in the fact that, the metal of the stay tubes being thicker than of the ordinary tubes, the former is more intensely heated. The stay tubes therefore expand more than the others, putting the latter in tension, the stay tubes being in compression. This is a reversion of the conditions existing when iron braces were used; these were put in tension and the tubes in compression. It would seem possible that, by choosing a material like iron or steel, having a coefficient of expansion less than that of copper, the stay tubes could be made of such a thickness that the total expansion would be identical with that of the ordinary tubes, and the strains mentioned above avoided. This plan has been tried, but no noticeable improvement has followed.

In order to determine a possible connection between the liability of tubes to leak and certain methods of construction, the author kept a record for a period of time of all the leaks occurring in the tubes of the six double-ended four-furnace boilers of a German ship, worked by artificial draft in closed fire-rooms.

The ordinary tubes were of brass, the stay tubes of iron. The number of leaks of corresponding tubes in all the forty-eight tube sheets, front and back, were transferred to a single tube sheet, and gave a diagram of the behavior of the different tubes magnified forty-eight times. It is apparent that such a proceeding will in great measure eliminate accidental defects due to faulty workmanship or material. Leakages in all tube sheets, front and back, were given the same weight, for the reason that any undue strain would make itself felt first where the tube was the less perfectly secured, *i. e.*, again in order to eliminate defects in workmanship.

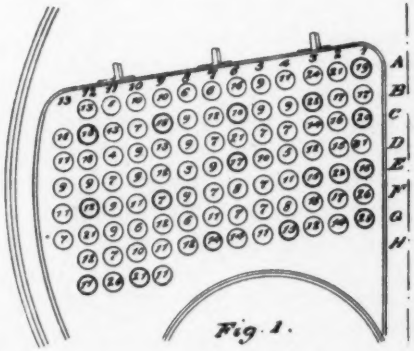

Fig. 1.

According to this record the 16 stay tubes for each tube sheet are credited with 262 leaks (48 times exaggerated), the 77 ordinary tubes with 908; or 16.4 for each stay tube, 11.8 for each ordinary tube. From this we conclude that something must have been wrong with the method of securing the stay tubes. They were secured in the back tube sheet by being screwed in, beaded over on the combustion chamber side and fitted with nut and washer on the water side. As the method of screwing into

tube sheets and beading has been successful elsewhere, the mischief must have been due to the nut and washer, and these we conclude are a source of weakness, not of strength, and should be left off. They produce overheating, as was proved in the present case, on inspection, by the reddish-gray color of the metal in the proximity of the nuts.

On turning to the figures shown on Fig. 1 we find:

*a.* For the 6 tubes immediately between two stay tubes ($B_1$, $C_6$, $D_1$, $F_1$, $E_2$, $H_1$), total number of leaks 143; for each tube, 23.9.

*b.* For the 3 tubes diagonally between two stay tubes ($D_2$, $F_2$, $F_8$), total leaks 38; for each tube, 12.7.

*c.* For 41 tubes immediately adjacent to the stay tubes, total number of leaks 506; for each tube, 12.3.

*d.* For 27 tubes diagonally adjacent to the stay tubes, total leaks 202; for each tube, 7.5.

These figures, which have been verified by observations on a number of ships, show that a compensation of the expansion of the tubes is impossible, and that stay tubes should not be used at all. Their omission involves no difficulties of construction, as the ordinary tubes are fully able to take the strain, and it is necessary only to see that the surfaces near the tube nests and clear spaces among the tubes are properly stayed.

Although the strain to which the tubes are subjected is low, yet brass is not a suitable material when the boiler is designed for forced draft.

Firstly, for the reason that the coefficient of expansion of brass is so much greater than that of the iron or steel of the adjoining braces and the shell, whose temperature, besides, is much lower.

Secondly, on account of the great diminution of strength on being heated. According to experiment the tensile strength of brass at 390° F. is only 73 per cent. of that at ordinary temperatures, at 570° only 50 per cent. and at 760° only 23 per cent. These considerations should preclude the use of brass, especially as the temperature to which it may be raised is uncer-

tain, being anywhere from that of the water, 320° to 380° F., to as high as 570° or even 1,000°.

Thirdly, on account of the softness of this material, and the fact that it loses on being heated what hardness and tenacity are imparted to it by rolling and expanding, resulting in the relaxation of the metal in the tube sheet, which destroys the tightness of the joint. Iron and steel, on the other hand, possess all those requirements of a good boiler material in which brass is found wanting, and have only the disadvantage that they are subject to corrosion. This corrosion, however, can be kept in check by proper care. Brass tubes and copper tube plates must therefore be regarded as things of the past.

To test these conclusions by actual practice we have only to refer to the locomotive boilers constructed by Schichau. These have without any doubt acquitted themselves under the ordeals of strong forced draft better than any other marine boiler. They are built entirely in accordance with the principles above laid down. The material is steel throughout; the tubes are all of uniform gauge; there are no stay tubes; the tubes are secured in the back tube sheet by being screwed in, expanded and beaded. Even these boilers, however, are not entirely free from occasional leakage of tubes, although these leaks are of comparatively rare occurrence and small extent. They are due, in the author's opinion, to the excessive closeness of spacing adopted. Overheating of the tubes near the tube sheets is likely to occur, and having occurred, any jar, as, for example, violent motion of the ship, concussion due to firing of a gun, sudden starting or stopping of the engine, may send the water in a rush to the overheated parts. The thin tubes cool off before the thicker tube sheet is affected, a rush of cold air in consequence of opening the fire door perhaps lends assistance, and a leakage results. With tubes spaced zigzag, overheating is more liable to occur than when the spacing is in horizontal and vertical rows, because the steam bubbles have no unobstructed path upward, but are constantly striking against and gliding along tubes. In such boilers the lower sides of the tubes are more strongly heated than the other parts, and as a consequence we have a curvature downward, which,

after the boiler has been in use for some time, is noticeable when no steam is up. This sagging amounts to about ½ inch with brass tubes about 8 feet long, and to about ¼ inch with steel tubes. While the boiler is in use this deflection may well amount to three times the above amount. Such a curvature is calculated to loosen the joint between the tube and tube sheet and produce leakage.

The overheating of the tubes near the tube sheet must, with strong forced draft, be very great, and the steam formed there very highly superheated; otherwise it would be impossible to explain why priming, which was such a common occurrence with the old-time box boilers, is a comparatively rare one now. Yet the steam volume produced in a given time in the boilers of to-day per square foot of water surface and per cubic foot of steam room is about what it was with the old boilers. The explanation is given by the assumption of the superheating of a portion of the steam formed, which in its turn gives off its heat to the moist steam and dries it. The overheating is evidenced by the peculiar characteristic color of the metal after a forced run. It is therefore necessary to provide for the free liberation of the steam bubbles as they are formed and the ready accessibility of the back tube sheet by the water by properly spacing the tubes. Schichau provides for this by opening out his vertical rows horizontally as the top is approached. This is a step in the right direction, but it is not sufficient. More space may be gained, either by increasing the horizontal distance between all the vertical rows, or by leaving an increased space between groups of vertical rows. The latter plan would appear to be the better one, as it accomplishes the object in view and at the same time permits of better utilizing the available space by employing zigzag spacing. As with increasing width of open spaces the strain thrown on the tubes is increased, we must limit the width of the spaces by this consideration. The number of vertical rows in each group should for this reason be from 2 to 4 or perhaps 5, not more, with intervals in proportion. The objections to be raised are increase in weight for the same heating surface, or else decrease of heating surface and decrease of calorimeter. As

regards the decrease of heating surface, the author is of the opinion that the decrease of the ratio of heating to grate surface in locomotive boilers to 45 : 1 would be accompanied by no loss of steaming capacity. In the cylindrical type of boiler, where this ratio is approximately 35 : 1, the loss of surface may be compensated for by decreasing the diameter of tubes. In proof of the assertion that any increase of the heating surface beyond 40 to 45 times the grate surface is without effect on the steam production, we may refer to column 9 of the table on page 353. Trials 5 and 7, using a ratio of heating surface to grate of 86.2, gave no greater evaporation than trials 1 and 4, when this ratio was 43.1. The author refers to another boiler, where the ratio was 68, and which gave no better results than other boilers with a smaller ratio; and he concludes that with natural draft there is no economic gain attached to increasing the ratio $\frac{\text{heating surface}}{\text{grate surface}}$ beyond 30, and in forced draft beyond 45. Any increase in weight contemplated would be better employed in widening the spacing of the tubes.

There is, besides, another way to facilitate the flow of steam through the tube nests, *i. e.*, introducing the feed water in the proper way. The practice of introducing it above the tube nests, near the back tube sheet, in the region where steam generation proceeds most violently, is much followed and is erroneous. For in such a case the colder entering water is heated at the expense of steam condensed, which has already laboriously worked its way up through the tubes. It is calculated that in a boiler carrying 180 lbs. pressure, from 15 to 20 per cent. in volume more steam must be formed when the feed enters as described than when it enters so that it may be heated directly by contact with heating surfaces or with hot water, *e. g.*, when brought in near the front tube sheet or at the sides of the furnace. The best practice of all is to pass the water through a heater before pumping into the boiler, and then to distribute it to make the best circulation.

If by this or any other means it becomes possible to avoid the accumulation of steam among the tubes and the consequent

superheating, it may be that with the disappearance of leaks the danger of priming will once more become prominent; on this head, however, actual experiment, not conjecture, can alone make a final answer.

The second of the dangers mentioned above as following in the train of forced draft is the splitting and tearing across of tubes. Such accidents are of rare occurrence, but may be attended with very serious consequences, if, the tubes tearing across, their weight weighs them down, leaving the openings exposed. Two cases of tubes breaking have come under the author's notice. One happened on a torpedo boat built by a German firm for a foreign power. The boat had run several trials when, while undergoing another, several tubes gave way close to the combustion chamber tube sheet. The tubes were of brass, screwed into the copper sheet (there being no stay tubes). The tubes, after removal, showed the reddish color peculiar to brass when overheated. They had parted, doubtless, after being weakened by the overheating. The same accident occurred on several of the boats, and the brass tubes were finally replaced by iron and steel ones.

The second similar accident occurred on a German naval vessel while steaming in the Mediterranean. The boilers were of the locomotive type. Twenty-nine tubes either split open or tore across near the tube sheet. In this case also examination showed evidence of overheating. The location of the tubes that failed, in relation to the stay tubes, is not known, but it seems just to conclude that the giving way of tubes is the ultimate result of the causes that are at the root of the leakage of tube ends above specified.

Another accident that brass boiler tubes are liable to is flattening. Two cases are instanced. The first was that of a vertical fire tube boiler: The tubes passed up through the steam space, and here two of the number were found to have been flattened, the deformation extending to the water level. Overheating, aided perhaps by slight irregularity in form was doubtless the cause. Iron tubes were substituted and gave no further trouble. The second case was that of a larger locomotive boiler with two fur-

naces. The boiler had successfully passed a number of trials, when finally, during an hour's forced run, so serious a leak developed that fires had to be hauled. It was discovered that 44 of the upper fire tubes had been, for lengths of from 8 to 20 inches, near the tube sheets, flattened out; one was found torn across. The flats were all in horizontal planes. Bulges in the top of the combustion chamber, leaky rivet seams and discolorations of the metal indicated that the water had been low. The overheating had, however, proceeded also from the accumulation of steam among the tubes, which were closely spaced, for tubes in the sixth row from the top were found flattened, and it is not to be supposed for an instant that the water level fell as low as that. The flattened tubes were found to have sagged about 2 inches, and it is the author's opinion that the action may have been started by this sagging, which was shown above to take place with forced working, gradually producing a nick near the tube sheet.

An evil of much more frequent occurrence than the last is the bulging of plates in contact with the fire. This evil antedates the application of forced draft, but has since its introduction been aggravated by insufficient water supply following upon the increased violence of steam formation. Scantiness of water supply is an evil very common in the water legs of locomotive boilers. These should be made of ample width, and circulation induced by artificial means. The sides of the fire boxes should never be vertical, but should be inclined, so that, even with the ship heeled over, steam will have a chance to rise without gliding along the surfaces. It is for the reason that the steam cannot leave the surface that, as stated above, it is poor practice to carry the grates of cylindrical furnaces below the middle. In cylindrical boilers of three or more furnaces the overlapping of the furnaces is objectionable for the same reason. Bulges may be due to excessive local heat, as in the case of a locomotive boiler whose bridgewall was inclined forward, forming a pocket with the grate. After a forced trial serious bulges were found to have developed in that locality. With the present high pressures and temperatures in boilers a bulge is quite a serious thing. The two sides of the

metal are of widely differing temperatures, and a powerful tendency exists to bend toward the fire side, which, exerting a powerful crushing strain on the metal near the stay bolts, must ultimately result in permanently weakening the plate. This crushing strain is greater the greater the thickness of the metal, the greater the coefficient of expansion, and the less the original strength. All these considerations go to show the inferiority of copper to steel for boiler furnaces, and the author states that his experience demonstrates the same, many more cracks having come to his notice in copper than in steel furnaces and tube sheets. With scarcely any exception these cracks extended between adjacent stay bolts or tubes. In this connection the author points out that copper tube sheets are easily deformed by the expanding of the tubes, unless drifts are driven into the holes surrounding the one into which the tube is being expanded. This expedient, however, requires much time, and does not avoid the injury done the metal acted on by the roller. With the steel furnaces of cylindrical boilers bulges are of rare occurrence, and when not due to overheating caused by accumulations of scale or greasy matter, they usually occur at the height of the grate, and are doubtless due to overheating brought about by the steam finding it impossible to leave the surface. This was the case with two boilers of a German ship that, after having been in use for 1½ years, suddenly developed, without being unduly pushed at the time, 55 bulges of varying extent and depth. The ship was encountering heavy weather, and it is supposed that the heel adding to the difficulties met with by the steam in making its escape from the furnace sides led to overheating.

Fracture of stay bolts is an evil confined to those in the water spaces surrounding the fire boxes of locomotive boilers. It is caused by the bending action produced by the inequality of expansion of the sheet in contact with the fire and the outer shell. These sheets being solidly united at the bottom, the majority of breaks should and do occur in the upper rows. While the bending action may lead to leaks at the inner sheet, the break always occurs at the inner surface of the outer sheet, because the stay bolt is relatively fixed there by the nut outside. Some

makers try to secure flexibility by forming at the outer sheet a species of ball-and-socket joint. It is a good plan to bore an axial hole into the stay bolt from the outside. Any break is then betrayed by the leak, which being easily plugged need cause no inconvenience. Because of the variable strains to which stay-bolts are subjected, they should be made only of the toughest material, steel or preferably iron.

Of the leakage of rivet seams nothing need be said, as the introduction of forced draft has developed nothing new in this direction.

In conclusion, the author speaks of a number of minor evils and their remedy. The rapid crumbling away of bridgewalls is obviated by moulding them solidly instead of building them up. The rapid burning away of the inner casing of front connection doors may be checked by attention to the length and diameter of tubes. Experiments made in Germany on the relative efficiency of long and short fire tubes of wide and narrow bore proved (1) that for equal grate surfaces long boilers with a given amount of heating surface are more effective than short boilers with more heating surface; (2) that with equal grate and heating surfaces larger tubes are somewhat less efficient than small. This result adds weight to the argument advanced above in favor of small tubes. For tubes of small diameter their length should not, for moderately forced draft, be less than 7.2 feet; for stronger forced draft not less than 8.2 feet; for tubes of large diameter these lengths should be 7.5 and 8.6 feet. In order not to fall into extremes it must be remembered, however, that the greatest length for moderate draft should not exceed 8.2 feet, for strong forced draft 9.8 feet, because any increase beyond these lengths either involves a sacrifice of other requisites or else adds weight comparatively useless.

It is doubtless one of the most difficult feats of designing to fill the available space in a ship with a boiler correctly proportioned in every way. It is not an impossible one, however, if the constructors of the hull and of the machinery make mutual concessions.

Among the numerous questions bearing on the subject under

discussion that is still waiting an answer based on practice and experience one of the most important is that concerning the air pressure permissible for different boiler types.

There is no doubt but that the wear and tear on a boiler increases with the quantity of fuel burnt per square foot of grate per hour; *i. e.*, with increase of air pressure. The latter cannot, however, be regarded as a just criterion of the degree of forcing a boiler is subjected to. For we have found most variable relations to exist between the air pressure carried and the coal burnt in boilers of identical construction. These discrepancies can be laid at the door only of either differences in the quality of coal or in the firing. The latter is the factor of chief importance, as the differences in the quality of coal are not great. For steam launch and locomotive boilers certain fixed relations seem to connect air pressure and coal consumption. These boilers permit of more regular firing than do cylindrical boilers, probably because of the fire doors being higher up and the grate more accessible. On a German ship, fitted with four double-ended boilers, two, containing sixteen furnaces, in each compartment, 31.05 tons were burnt in one compartment during the 6 hours' trial on 1.81 inches of water, while 29.25 tons were burnt in the other on .98 inch. Difference in quality of coal is out of the question, it is entirely a matter of management of fires. To burn 31.05 tons should have required a pressure of $\left(\frac{31.05}{29.25}\right)^2 \times .98 =$ 1.08 inches; or, with 1.81 inches at least $29.25 \times \sqrt{\frac{1.81}{.98}} = 39.8$ tons should have been burned. The firing was much more skilfully done in one compartment than in the other. Great practice is required to keep the coal in a cramped cylindrical furnace with low fire door and 6.5 feet or more in length perfectly level. Another point of great importance is that of keeping all the fires in a compartment, in the present case sixteen with a total surface of 323 square feet, at the same thickness. If a uniform thickness is not maintained, some of the fires will burn out rapidly, others will be inert. The aim should be, in grouping the boilers, to get them distributed into as many separate compartments as possible,

making the grate surface in each from 32 to 43 square feet, or, at any rate, not more than 64 to 86.

So far it is impossible to fix positively on the upper limit of air pressure to be carried with any given boiler, without, perhaps, curtailing its power unnecessarily or even excessively; but it should be possible so to construct boilers that—proper firing being presupposed—launch boilers and locomotive boilers of moderate size could burn 50 to 60 lbs. per square foot of grate with a maximum pressure of 2.36 inches of water; large locomotive boilers (containing several furnaces) could burn 40 to 50 lbs. on a maximum of 2 inches; cylindrical boilers could burn from 26 to 30 lbs. per hour.

To judge properly of the firing, it is necessary to know more than the pressure in the fire room. The pressure difference between fire room and furnace should be known, and the information is complete only when the pressure existing in the combustion chamber and the smoke box is known likewise. During the trials of the German ships referred to above these quantities were measured, and the following are the means of 7 readings:

| | | |
|---|---|---|
| Pressure in after fire room, | | .83 inch. |
| Pressure in furnace, | | .31 inch. |
| Pressure in corresponding combustion chamber, | (—) | .08 inch. |
| Pressure in corresponding smoke box, | (—) | .55 inch. |

It is evident from these figures that the blowers and the chimney draft shared the work, so that in the present case it would be more proper to speak of assisted than of forced draft. A point to be remarked is that less pressure was required to force the air through the grate and the fuel than through the opening over the bridgewall.

The systematic recording of such data on trials in connection with the measurement of the coal consumed in each compartment or by each boiler, notice being taken of individual differences in the method of firing, might lead to some valuable results, and perhaps furnish a basis for determining the proper limit of air pressure for a boiler.

## XXIII.

# MARINE ENGINE TRIALS.

[Research Committee on Marine Engine Trials. Report upon trials of *S. S. Iona.*]

By Professor Alexander B. W. Kennedy, F. R. S., Chairman.

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Since their report of May, 1890, the Research Committee on Marine Engine Trials have had the opportunity of carrying out another trial, which will be found in some respects more complete than any of those they have made previously.

*Steamer.*—The steamer tested was the *Iona*, owned by Messrs. Herskind and Woods, of West Hartlepool. The *Iona* is a fine example of a modern well-decked cargo schooner with economical engines. She was built in 1889 by Messrs. W. Gray & Co., of West Hartlepool, and engined by the Central Marine Engine Works of the same firm with triple-expansion engines on three cranks working a single screw. She is a vessel of 275.1 ft. length, 37.3 ft. breadth, and 19 ft. depth. Her depth moulded is 21 ft. 10 in. and her coefficient of fineness is 0.765. She has a double bottom of cellular construction 230 ft. long, and can carry 443 tons of water ballast. Her registered tonnage is—gross 2,094 tons, under deck 1,610 tons, and net 1,348 tons, and she is classed at Lloyd's 100 A 1 (special survey). Her draft in Tyne Docks before the trial started was 20 ft. 7 in. forward, and 20 ft. 9 in. aft, or 20 ft. 8 in. mean, but in salt water she rose ½ in. as measured by hydrometer, so that the mean draft during the trial was 20 ft. 7½ in., corresponding to a displacement of 4,430 tons.

The *Iona* arrived at Middlesbrough from her voyage on June 23, 1890, and her engines were afterwards overhauled by their makers. She subsequently went north to the Tyne to take in a cargo of coal, and the trial was made during her outward voyage south. The steamer left Tyne Docks at 12.20 noon on July 13, 1890, and crossed the bar about 1 o'clock, the weather being fine

with a fresh breeze. The fires had been lit since 6 A. M. on the previous day, and in order to get everything warmed up and the fires in normal condition, the trial itself was not started until 6.30 P. M. on 13th July, by which time the steamer was off Robin Hood Bay. The trial lasted for 16 hours, ending at 10.30 A. M., on 14th July, when the steamer was off Yarmouth. The weather remained fine throughout, so that there was no difficulty in taking any of the usual observations. The speed of the engines was not altered during the trial, nor was the valve gear touched or altered throughout the whole time. In addition to all the observations which had been taken on former trials, indicator diagrams were taken at intervals from the circulating and air pumps, from the first and second receivers, and from both ends of the condenser. The temperatures of the circulating water and discharge water were also noted; and the jacket water, steam pipe drain water, and supplementary feed water were each separately measured.

*Engines.*—The *Iona* is fitted with triple-expansion surface-condensing engines, the cylinders being placed in the order—intermediate, high, low, going from forward to aft. The cranks rotate in the sequence—high, intermediate, low. The diameters of the cylinders are 21.88 in., 34.02 in. and 56.95 in., by gauges. The piston rods are all 5 in. in diameter. There are no tail rods. The stroke of all three cylinders is 39 in. The high pressure cylinder only is jacketed. Its jacket is supplied with steam from the boiler direct; under ordinary working conditions it is drained directly into the hot well, but during the trial it was drained through a measuring tank. A drain pipe was taken from the steam pipe, and another from the high pressure valve chest, and these were connected to another tank, so as to be also separately measured. The intermediate and low pressure cylinders are not jacketed. The clearance volumes of the cylinders are given by the makers as 12.41, 10.11 and 7.64 per cent. of the volumes swept through by their respective pistons.

The high-pressure and low-pressure cylinders are provided with ordinary double-ported slide valves, and the intermediate cylinder with a Trick slide valve. None of the slide valves are

balanced. The valve gear is the ordinary link motion. There are two feed pumps, one circulating pump, one air pump, and two bilge pumps, all driven by levers from the crosshead of the low-pressure cylinder. The feed pumps are each 3 in. in diameter, single-acting; the circulating pump is 10 in. in diameter, double-acting; the air pump is 17 in. in diameter, single-acting, and the bilge pumps are 3½ in. in diameter, single-acting. The stroke of all six pumps is 26 in. The surface condenser contains 1,360 square feet of tube surface, in 475 tubes of ¾-in. external diameter and 14 ft. 7 in. length between tube plates. The screw propeller is of cast iron, having four blades, and is 14 ft. 6 in. diameter, with a mean pitch of 14 ft. 6 in. and a surface of 63 square feet.

*Boilers.*—Steam is supplied by two single-ended steel boilers with Purves' corrugated flues. The boilers are 13 ft. 3 in. in mean diameter and 10 ft. long, with two furnaces each, or four furnaces in all. The total grate area was originally 52.5 square feet, each grate being 3 ft. 9 in. long and 3 ft. 6 in broad; but the fire bars have since been shortened by the length of one brick to 3 ft., so that at the time of the trial the total grate area was only 42 square feet. The total heating surface of the two boilers is 3,160 square feet, of which 2,590 square feet is tube surface. The total heating surface is therefore 75.2 times, and the tube surface 61.7 times the actual grate area. The ratios of total heating surface and tube surface to the original grate area are 60.2 and 49.3 respectively. There are in all 480 tubes, 3 in. external diameter and 6 ft. 10⅝ in. long between plates. The internal diameter of the funnel is 6 ft. 3 in., and its total height is 51 ft. 9 in. above the level of the fire grates. A damper is placed in the funnel at a height of 17 ft. 8 in. above the level of the fire grates. The total cross-sectional area through tubes is 18.3 square feet, and the area across the funnel is 30.7 square feet. The net volume of the boilers is 2,680 cubic feet.

The boilers are worked under forced draft, on a plan designed by Mr. John R. Fothergill. A fan in the engine room, driven by an engine which during the trial received steam from the donkey

boiler, forces air into a trunk which extends across the front of the boilers. This air passes into the furnaces through gridiron valves placed just below the level of the fire bars; each of these valves is made to close automatically by the lifting of the latch whenever the corresponding furnace door is opened, thus preventing flame and gases from being blown out from the furnace into the stokehold. A smaller air trunk, branching off from the main trunk and provided with a deflecting valve, passes along the back of the boilers and admits a quantity of air through perforated plates into the combustion chamber, in the form of small streams which mix with the gases as they pass the furnace bridge; the mixture is then deflected to the bottom of the combustion chamber by a plate hung at the back end of the furnace. The fan is 60 in. in diameter and 20 in. broad, and is capable of supplying 1,400 cubic feet of air per minute at a pressure of $3\frac{1}{2}$ in. water gauge; it is driven direct by a Chandler's single-acting high-speed engine, with cylinder 8 in. in diameter and 7 in. stroke, developing 2.1 indicated horse-power at a speed of 180 revolutions per minute, which was the mean speed during the trial. The fan engine is usually supplied with steam from the first receiver of the main engines; but throughout the trial, as already stated, steam was supplied to it from the donkey boiler. Water gauges connected to the casings and ash-pits were read at intervals of half an hour, and the following are their mean readings:

| Position of Gauge. | Pressure. Inch of Water. |
|---|---|
| On air trunk near fan, | 0.86 |
| On air trunk at back of boilers, | 0.34 |
| On ash-pits, mean of four gauges, | 0.17 |

*Weights.*—The total weight of the engines and boilers and all mountings, including water in condenser, pipes, and boilers is about 202 tons, exclusive of shafting, tunnel bearings and propeller. The following are the weights of the various parts of the machinery, boilers and accessories, as given by the makers:

| | Tons. |
|---|---|
| Engines alone, | 64.92 |
| Shafting, tunnel bearings and propeller, | 26.59 |
| Engine-room auxiliaries, including donkeys, pipes, platforms, ladders and gratings, | 12.16 |
| Water in condenser, pipes and pumps, | 1.71 |
| Boilers alone, | 58.60 |
| Boiler-room auxiliaries, including forced draft gear, smokebox, uptake, funnel, furnace gear, mountings, stokehold floor, boiler chocks and ties. | 28.49 |
| Water in boilers, | 35.75 |
| Total, | 228.22 |

*Duration of trial.*—The duration of the trial from start to finish was exactly 16 hours, or 960 minutes.

*Coal measurement.*—The coal was weighed in baskets by means of a spring balance in the stokehold, about 450 lb. at a time being put down on the floor beside each boiler. The trial was started with clean floors, and the time of first stoking from each weighed lot of coal was noted, as well as the time when each weighed lot was completely put on the grates. The coal consumption plots out as shown in Fig. 1. As an additional check upon the coal measurement a particular bunker was emptied beforehand, and coal enough for the trial was weighed into it off the trucks. The amount of coal so weighed amounted to 9.61 tons. At the end of the trial the quantity of coal still left in the bunker was taken out and weighed, and amounted to 2.82 tons, which, deducted from the whole quantity weighed into the bunker, gives 6.79 tons as the quantity used. As the coal weighed in detail during the trial was 6.67 tons, the difference between the two methods of measurement was less than 1¼ per cent. of the whole quantity in the bunker.

The fires were cleaned once during the trial. The total amount of ash, weighed after the trial, amounted to 430 lb., which is about 2.9 per cent. of the whole fuel used.

The coal used throughout was Tyne coal from Walbottle Colliery. Coal samples were taken from the weighed lots frequently

during the trial; and after a thorough mixture of all these samples, the final analyses, which have again been kindly made by Mr. C. J. Wilson, are as follows:

| | Coal as Used. Per cent. | Dry Coal. Per cent. |
|---|---|---|
| Carbon, . . . . . | 82.34 | 83.97 |
| Hydrogen, . . . . | 5.47 | 5.58 |
| Moisture, . . . . . | 1.94 | 0.00 |
| Ash, . . . . . | 2.90 | 2.96 |
| Nitrogen, sulphur, oxygen, &c., by difference . . . | 7.35 | 7.49 |
| | 100.00 | 100.00 |

A sample of the ash was also collected, and has been analyzed by Mr. Wilson as follows:

| | Per cent. |
|---|---|
| Loss on ignition (=carbon), . . . | 56.19 |
| Mineral matter, . . . . . | 43.81 |
| | 100.00 |

The calculated calorific value of the fuel is 14,830 thermal units per pound, which corresponds to the evaporation of 15.35 lb. of water from and at 212 deg. Fahr., and to an equivalent carbon value of 1.02 lb. per pound of coal. A portion of the coal sample was also tested by means of a Thompson calorimeter of modified form, which gave a calorific value of 14,890 thermal units per pound as the mean of two separate determinations; the difference between this and the value calculated from the analysis is less than half of one per cent. The total coal used was as follows:

| | lb. |
|---|---|
| Port boiler, . . . . . | 7,248 |
| Starboard boiler, . . . . | 7,701 |
| | 14,949 |

This amounts to 15.7 lb. per minute, or 942 lb. per hour, taking a total time for coal consumption of 15 hours 53 minutes, or 953 minutes, which was arrived at as follows: The last firing with unweighed coal before the trial began was at 6.25 P. M. upon July

13, and the last firing with weighed coal during the trial was at 10.21 A. M. upon the 14th, or an interval of 956 minutes. The first firing with weighed coal during the trial was at 6.38 P. M. upon the 13th, and the weighed coal was all burnt at 10.28 P. M. upon the 14th, which was the time of first firing with unweighed coal after the trial was over, or an interval of 950 minutes. A mean between these two intervals, or 953 minutes, has therefore been taken as the total time for coal consumption.

*Furnace Gases.*—Thirteen samples of the furnace gases were obtained, being collected over mercury by Mr. Wilson as the trial went on. Some of these, it will be seen, were collected quickly, as for instance Nos. 5 and 10. The others were collected slowly over periods varying from half an hour to an hour. The differences between the specimens collected quickly and slowly, it will be seen, are not at all so great as those between different samples collected at about the same rate. The analyses of the samples are given in Table I.

The chimney temperatures were read every half hour with a mercury thermometer. Two Murrie pyrometers were also used alongside of the thermometer to indicate the chimney temperatures, and the readings of the three instruments agreed. The height of the thermometer above the furnace bars was about 30 ft. The average reading was 452 deg. Fahr., the minimum being 410 deg., and the maximum 478 deg. The variations in chimney temperature are shown in Fig. 1. Trial was made to see whether any alteration of temperature reading followed a change in the position of the thermometer bulb from the side to the center of the chimney, but none could be detected. The mean air temperature on deck was 62 deg. Fahr.

The mean chimney draft, which was measured by a U gauge-glass at the place where the furnace gases were collected, or about 13 ft. above the damper, was 0.25 in. of water. The funnel damper was only one-sixth open throughout the whole of the trial.

*Feed-Water Measurement.*—The feed water was measured in the tanks which had been previously used in the *Colchester* and the *Tartar* trials. In the present trial each tank lasted about

TABLE I.—*Iona* TRIAL.

| Number of Sample. | Analyses of Funnel Gases by Volume. | | | | | Analyses of Funnel Gases by Weight. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Carbonic Acid. | Carbonic Oxide. | Oxygen. | Nitrogen. | Time of Collecting. | Duration of Collecting. | Carbonic Acid. | Carbonic Oxide. | Oxygen. | Nitrogen. |
| | per cent. | per cent. | per cent. | per cent. | P. M. | mins. | per cent. | per cent. | per cent. | per cent. |
| 1 | 6.43 | 0.00 | 13.11 | 80.46 | 6.34– 7.20 | 46 | 9.57 | 0.00 | 14.20 | 76.23 |
| 2 | 8.71 | 0.00 | 10.51 | 80.78 | 7.30– 8.10 | 40 | 12.85 | 0.00 | 11.28 | 75.87 |
| 3 | 8.64 | 0.00 | 10.61 | 80.75 | 8.30– 9.30 | 60 | 12.75 | 0.00 | 11.39 | 75.86 |
| 4 | 7.79 | 0.00 | 11.63 | 80.58 | 9.30–10.30 | 60 | 11.54 | 0.00 | 12.52 | 75.94 |
| 5 | 8.86 | 0.00 | 10.79 | 80.35 | 10.30 A. M. | 2 | 13.06 | 0.00 | 11.57 | 75.37 |
| * | | | | | | | | | | |
| 9 | 8.93 | 0.00 | 9.67 | 81.40 | 2.30– 3.20 | 50 | 13.18 | 0.00 | 10.38 | 76.44 |
| 10 | 8.15 | 0.00 | 11.17 | 80.68 | 3.30 | 2 | 12.05 | 0.00 | 12.01 | 75.94 |
| 11 | 7.15 | 0.00 | 11.70 | 81.15 | 4.30– 5.30 | 60 | 10.62 | 0.00 | 12.64 | 76.74 |
| 12 | 7.06 | 0.00 | 12.51 | 80.43 | 5.30– 6.30 | 60 | 10.48 | 0.00 | 13.51 | 76.01 |
| 13 | 7.91 | 0.00 | 10.80 | 80.29 | 6.30– 7.30 | 60 | 13.13 | 0.00 | 11.57 | 75.30 |
| 14 | 8.85 | 0.00 | 10.87 | 80.28 | 8.00– 8.30 | 30 | 13.05 | 0.00 | 11.65 | 75.30 |
| 15 | 8.82 | 0.00 | 10.78 | 80.40 | 9.00– 9.30 | 30 | 13.00 | 0.00 | 11.56 | 75.44 |
| 16 | 8.28 | 0.00 | 11.07 | 80.65 | 9.40–10.15 | 35 | 12.24 | 0.00 | 11.90 | 75.86 |
| Mean ... | 8.20 | 0.00 | 11.17 | 80.63 | ..... | ..... | 12.12 | 0.00 | 12.01 | 75.87 |

*Samples 6, 7 and 8 were not properly taken, and were therefore rejected as being unreliable.

15½ minutes. As already mentioned, separate tanks were used for measuring the jacket water, and the water from the steam pipe and high-pressure valve chest drains, and a third separate tank was used for measuring the supplementary feed water. The water was pumped from the hot well into the measuring tanks by a donkey pump which was supplied with steam from the donkey boiler. This arrangement was adopted in order to prevent the additional load which would have been thrown upon the air pump if it had had to deliver into the measuring tanks. The feeding of the main boilers was done entirely by the engine pumps, which drew the water from the measuring tanks. The glands of the feed pumps leaked considerably during the first hour and a half of the trial; but the leakage was then so far stopped that it could be neglected during the rest of the trial.

Steam was kept up in the donkey boiler for the fan engine, the donkey pump already mentioned, and the ballast donkey.

The total amount of water used was 136,820 lb. during a total time of 953 minutes. This amounts to 143.6 lb. per minute, or 8,616 lb. per hour. The rate at which the feed water was supplied, and the variations of its temperature, are shown in Fig. 1.

Both at the commencement and at the end of the trial the boiler pressure was falling, as will be noticed from the diagram, Fig. 3, and the water level was as nearly as possible the same; but the motion of the vessel at the end of the trial made the last observation of water level a little uncertain. An error of ½ in., however, which is certainly more than could have occurred, would make a difference of less than 0.4 per cent. in the total feed, and might therefore be practically neglected.

At 5 A. M. on July 14 samples of water were taken from both boilers, and at the same time a quantity of steam from the main steam pipe was condensed and collected. These samples were analyzed by Mr. Wilson to determine the percentage of salt present in each, and gave the following results:

| Sample. | Percentage of Salt present. |
|---|---|
| From steam pipe, . . . | 0.00758 per cent. |
| From starboard boiler, . . | 0.351 " |
| From port " . . | 0.177 " |
| Mean of both boilers, . . | 0.264 " |

Taking the average saltness of the water in the two boilers, this shows that the above sample of condensed steam contained 2.87 per cent. of boiler water, which figure therefore represents the amount of priming at the time when the sample was collected. It was intended to have taken a series of these samples throughout the trial; but unfortunately the condensing apparatus broke down, and the above sample was the only reliable one that was obtained.

The temperatures of the circulating water and of the discharge water were measured at about half-hourly intervals throughout the trial. Calculating from these observations, the mean temperature of the circulating water was 55.8 deg. Fahr., and of the discharge water 75.5 deg. Fahr.

*Additional Water Measurements.*—The total water taken from the high-pressure jacket over a period of 910 minutes was 5,600 lb., being at the rate of 6.15 lb. per minute, or 4.3 per cent. of the total feed.

The total water from the steam pipe and high-pressure valve chest drains during 910 minutes was 800 lb., or 0.88 lb. per minute, which is 0.61 per cent. of the total feed.

The total amount of supplementary water was 8,302 lb. during 953 minutes, being 8.7 lb. per minute, or 6.1 per cent. of the total feed.

Practically the whole of the water from the jacket and cylinder drains was returned into the feed-measuring tanks, and therefore repumped into the boilers.

*Power Measurement.*—Indicator diagrams were taken every half hour from each end of each cylinder, as in the former trials. There were thus 32 complete sets of diagrams, or 192 single diagrams, taken in all. The indicators used were again kindly lent by Messrs. T. S. McInnes & Co., of Glasgow, and were the same that had been in use in the trials of the *Colchester* and *Tartar*. The springs used were in most cases new, and had all been carefully tested before the trial. The following are the mean effective pressures in the cylinders in pounds per square inch:

| Cylinder. | Top. | Bottom. | Mean. |
|---|---|---|---|
| High-pressure, . . | 47.98 | 45.32 | 46.65 |
| Intermediate, . . | 20.52 | 20.36 | 20.44 |
| Low-pressure, . . | 6.80 | 7.52 | 7.16 |

These pressures correspond with the following indicated horse-powers:

| | |
|---|---|
| High-pressure cylinder, . . . | 205.6 |
| Intermediate " . . . | 221.2 |
| Low pressure " . . . . | 218.6 |
| Total indicated horse-power, . . | 645.4 |

The maximum indicated horse-power given by any one set of diagrams was 667.9, which occurred at 8.15 A. M. on July 14, with 61.3 revolutions per minute, and boiler pressure of 163 lb. per square inch. The minimum indicated horse-power by any one set of diagrams was 627.5, at 7.15 P. M. on July 13, with 61.4 revolutions per minute, and boiler pressure of 162 lb. per square inch. In Figs. 6 to 8 is given the set of diagrams nearest to the mean, which was taken at 10.45 P. M. on July 13, with 61.5 revolutions per minute, and boiler pressure of 165 lb. per square inch. Eight diagrams were taken from the circulating pump during the trial, four from each end, and examples of these are shown in Fig. 13. The average power given by these diagrams is 12.4 indicated horse-power at the mean speed of 61.1 revolutions per minute. Six diagrams were taken from the air pump, which is single-acting, and examples of these are shown in Fig. 12. The average power given by these diagrams is 1.2 indicated horse-power at the mean speed of 61.1 revolutions per minute. Several diagrams were also taken during the trial from the first receiver, the indicator barrel being actuated from the crosshead of the intermediate cylinder; from the second receiver when the barrel was actuated from the low-pressure crosshead, and again when it was actuated from the high-pressure crosshead; and also from both ends of the condenser. Examples of the receiver diagrams are shown in Figs. 9, 10 and 11. The condenser diagrams are merely horizontal straight lines, showing a mean absolute back pressure of 1.2 lb. per square inch. The continuous variations of boiler pressure are shown in Fig. 3, of cylinder mean effective pressures in Fig. 5, of horse-power in Fig. 2, and of speed in Fig. 4.

*Speed.*—The counter was read every half-hour. The total number of revolutions made was 58,651 in sixteen hours, giving an average speed of 61.1 revolutions per minute throughout the trial. The maximum number of revolutions per minute for any half-hour during the trial was 61.8, and the minimum 59.7 revolutions. The continuous increase of total revolutions from the beginning of the trial is shown in Fig. 1.

*Pressures, &c.*—All the pressure gauges connected with the engines and boilers were checked by means of a standard gauge, and a correction allowed for where necessary. The mean barometric pressure during the trial was 29.67 in. of mercury, or 14.58 lb. per square inch. The mean boiler pressure was 165.0 lb. per square inch above the atmosphere by the pressure gauges in the engine room. The mean pressures in the high-pressure valve chest, first receiver and second receiver, were respectively 160.0 lb., 41.7 lb. and 2.4 lb. per square inch above the atmosphere. The mean vacuum in the condenser was 28.25 in. of mercury by gauge, which corresponds with a mean absolute back pressure in the condenser of 0.70 lb. per square inch. The mean pressures in the first and second receivers, as measured from the indicator diagrams taken, were respectively 38.0 lb. and 0.6 lb. per square inch above the atmosphere, and the mean absolute back pressure in the condenser, as measured from the diagrams, was 1.0 forward and 1.4 aft, or a mean of 1.2 lb. per square inch. The mean initial pressure in the high-pressure cylinder, as measured from the diagrams, was 142.5 lb. per square inch above the atmosphere; and the mean vacuum in the low-pressure cylinder, as measured from the diagrams, was 12.74 lb. per square inch, which corresponds with a mean absolute back pressure of 1.84 lb. per square inch in the cylinder.

*Boiler Efficiencies.*—The mean rate of combustion in the furnaces was 22.4 lb. of coal per square foot of actual grate surface per hour, or 0.30 lb. per square foot of total heating surface per hour. The total amount of feed water pumped into the boilers, as given in an earlier paragraph, corresponds with 9.15 lb. per pound of coal. The average temperature of the feed was 106 deg. Fahr., and the temperature corresponding with the mean

boiler pressure was 373 deg. Fah., so that each pound of steam must have taken up $(212-106)+966+0.305(373-212)=1,122$ thermal units. The equivalent evaporation from and at 212 deg. Fahr. was therefore 10.63 lb. of water per pound of coal, or 10.42 lb. per pound of carbon value in the coal. The equivalent amount of heat utilized per pound of coal would be 10,265 thermal units, or 69.2 per cent. of the thermal value of the fuel. This figure, of course, represents the actual boiler efficiency.

The series of analyses of chimney gases enable the weight of the air used per pound of coal to be calculated. The weight of dry air per pound of coal thus worked out amounts to 24.5 lb., so that the total weight of furnace gases per pound of coal would be about 25.4 lb. These gases have been raised in temperature from 62 deg. Fahr., the temperature of the outer air, to 452 deg. Fahr., the mean chimney temperature. Taking the mean specific heat of the gases as 0.243, this corresponds to a loss of 2,407 thermal units per pound of coal, or 16.2 per cent. of the whole calorific value of the fuel. In estimating the weight and specific heat of the gases, it has been assumed that the whole of the hydrogen in the fuel was burnt. No doubt this is not an entirely correct assumption, but the error due to it is very small. The loss due to the evaporation of the moisture in the fuel is so small that it may be neglected; and in no one of the thirteen analyses was any carbonic oxide found to be present in the gases. The quantity of heat not accounted for is therefore 14.6 per cent., which is mainly covered by the radiation, but includes also any losses due to unburnt carbon passing up the chimney, and to imperfect combustion of the hydrocarbons, &c.

The weight of air theoretically necessary for the complete combustion of 1 lb. of coal is 11.4 lb. The air actually used is therefore about 2.15 times the amount theoretically required.

The weight of water evaporated from and at 212 deg. Fahr. per square foot of total heating surface was 3.17 lb. per hour.

The average rate of transmission of heat through the material of the boiler was 3,059 thermal units per square foot of heating surface per hour.

*Coal Consumption.*—The total coal burnt, namely 942 lb. per

hour, corresponds to 1.46 lb. per indicated horse-power per hour. This is equivalent to 1.49 lb. of carbon value per indicated horse-power per hour.

*Engine Efficiencies.*—The total indicated horse-power being 645.4, and the total feed water used per hour being 8,616 lb., the feed water used per indicated horse-power per hour was 13.35 lb. The actual heat received by the feed water per minute was 161,100 thermal units, or 249.6 thermal units per indicated horse-power per minute, which is 69.2 per cent of the whole heat of combustion. The absolute engine efficiency, or ratio of the heat turned into work to the heat given to the feed water was 17.1 per cent.

*Total Efficiency.*—The combined efficiency of the boilers and engines, or ratio of the heat turned into work to the total heat of combustion of the fuel, was $0.692 \times 0.171$, which is equivalent to 11.8 per cent.

*Steam from Indicator Diagrams.*—The following are the results of measurements made upon the indicator diagrams taken to ascertain the proportion of steam accounted for by them. The actual weight of feed water used per revolution was 2.35 lb.:

| Proportion of steam accounted for by indicator diagrams. | Pounds per revolution. | Percentage of total feed. | Percentage in jacket or present in cylinder as water. |
|---|---|---|---|
| | lb. | per cent. | per cent. |
| Steam present in high pressure cylinder after cut off, when the pressure was 125.4 lb. per square inch above the atmosphere..... ..... ..... ..... | 1.49 | 63.4 | 33.6 |
| Steam present in intermediate cylinder, when the pressure was 19.4 lb. per square inch above the atmosphere..... | 1.76 | 74.9 | 25.1 |
| Steam present in low pressure cylinder near end of expansion, when the pressure was 8.6 lb. per square inch below the atmosphere..... ..... ..... | 1.39 | 59.1 | 40.9 |

*Subsequent Trial by Chief Engineer.*—After the trial above described had been carried out, the *Iona* made a voyage up the

Mediterranean, and on her return journey from the island of Serpho to Rotterdam, on August 15, 1890, when the vessel was off the coasts of Tunis and Algeria, the chief engineer of the vessel, Mr. J. Fiddes Brown, carried out a trial on his own account. The weather was fine throughout the trial, with light head winds, but the vessel was rolling somewhat during the latter part. The mean draft during the trial was about 20 ft. 9 in.

The trial commenced at 5.33 A. M., and was continued until 4.26 P. M., by chronometer, thus lasting for 10 hours 53 minutes, or 653 minutes.

As in the Committee's trial in July, the coal was twice weighed, the difference between the two methods of weighing being only 1.2 per cent., which is accounted for by the difficulty of weighing accurately during the rolling in the latter part of the trial. The total coal used was 10,631 lb., or 16.3 lb. per minute.

Indicator diagrams were taken from each end of each cylinder every half hour throughout the whole trial. Two Richards' indicators were used; one was kept constantly on the high pressure cylinder, while the other was changed between the intermediate and low pressure cylinders. The indicators were attached to the cylinders by the usual long pipes and bends, and top and bottom diagrams from the same cylinder were taken on each paper. In this way 22 complete sets, or 66 separate papers with two diagrams on each, were obtained. The maximum indicated horse-power from any one set of diagrams was 730.0, taken at 1 P. M., and the minimum from any one set was 683.5, taken at 3.30 P. M.

The revolutions were counted once in each quarter of an hour, and the mean of all the observations gave an average speed throughout the trial of 63.9 revolutions per minute.

The mean barometric pressure throughout the trial was 30.10 in. of mercury, or 14.8 lb. per square inch. The mean pressures in the first and second receivers were respectively 41.5 lb. and 1.8 lb. per square inch above the atmosphere. The mean readings of the water gauges connected with the forced-draft apparatus were as follows:

| Position of Gauge. | Pressure. |
|---|---|
| On air trunk near fan, . . | 1.03 in. of water. |
| On air trunk at back of boilers, . | 0.78 " " |
| On ash-pits, mean of four gauges, | 0.32 " " |

The mean speed of the fan engine was 218.5 revolutions per minute, and the indicated horse-power developed was 2.6. As in the Committee's trial in July, steam was supplied to the fan engine from the donkey boiler. The mean temperature of the gases in the uptake of the furnace, as indicated by a pyrometer placed at a height of 6½ ft. above the level of the firebars, is given as 672 degs. Fahr.

The principal results of the August trial are given in Table II, along with those of the Committee's trial, and it will be seen that these two sets of results agree remarkably well. The only essential difference between them is that in the August trial the coal is given as 1.38 lb. per indicated horse-power per hour, whereas in the July trial it was 1.46 lb. per indicated horse-power per hour. The same description of coal was used in both of the trials. In the August trial the feed water was not measured, and the drain and jacket water was delivered into the hot-well, instead of being allowed to collect gradually in measuring tanks before being mixed with the feed water. In consequence mainly of this, the feed water entered the boiler at a mean temperature of 157 deg. Fahr., whereas in the July trial the mean feed temperature was 106 deg. Fahr. The difference between the amount of heat taken up in the boiler by 1 lb. of water in the two trials is 4.6 per cent. of the larger amount. The difference between the two fuel consumptions is 5.5 per cent. of the larger amount. These two results strikingly corroborate each other, and show how exactly it is possible to estimate beforehand the actual saving due to the heating of the feed water. The consumption of coal in the August trial, if calculated for the conditions under which the July trial was carried out, would be 1.45 lb. of coal per indicated horse-power per hour.

*Speed of Vessel.*—Besides careful measurements from the ship's chart, taken by Captain Cranston at regular intervals throughout the Committee's trial and embodied in the ship's log, a continu-

ous registering log was employed to estimate the speed of the vessel, and the reading was noted every half hour during the trial by the captain.

TABLE II.—COMPARATIVE RESULTS OF TWO TRIALS OF THE STEAMER *Iona*.

| | | | |
|---|---|---|---|
| 1 | Trial made by | Research Committee. | Chief Engineer. |
| 2 | Date of trial | July 13& 14, 1890. | Aug. 15, 1890. |
| 3 | Duration of trial ..... hrs... | 16 | 10.88 |
| 4 | Type of engines | Triple. | Triple. |
| 5 | Cylinder diameter, H.P. ..... in... | J 21.88 | J 21.88 |
| 6 | Cylinder diameter, I.P. ..... in... | N 34.02 | N 34.02 |
| 7 | Cylinder diameter, L.P. ..... in... | N 56.95 | N 56.95 |
| 8 | Stroke, length ..... in... | 39 | 39 |
| 9 | Boilers, number of main boilers | 2 | 2 |
| 10 | Boilers, single-ended or double-ended | Single. | Single. |
| 11 | Furnaces, total number | 4 | 4 |
| 12 | Heating surface, total ..... sq. ft... | 3,160 | 3,160 |
| 13 | Heating surface, tubes ..... sq. ft... | 2,590 | 2,590 |
| 14 | Grate area (actual) ..... sq. ft... | 42 | 42 |
| 15 | Total heating surface to grate area ..... ratio... | 75.2 | 75.2 |
| 16 | Tube surface to grate area ..... ratio... | 61.7 | 61.7 |
| 17 | Grate area to flue area through tubes ..... ratio... | 2.3 | 2.3 |
| 18 | Grate area to area through funuel ..... ratio... | 1 4 | 1.4 |
| | *Mean Pressures.* | | |
| 19 | Mean boiler pressure above atmosphere ..... lb. per sq. in... | 165.0 | 162.0 |
| 20 | Mean admission pressure, H.P. cylinder above atmosphere ..... lb. per sq. in... | 142.5 | 147.8 |
| 21 | Mean effective pressure H.P. cylinder ..... lb. per sq. in.. | 46.65 | 47.33 |
| 22 | Mean effective pressure I P. cylinder ..... lb. per sq. in... | 20.44 | 20.21 |
| 23 | Mean effective pressure L.P. cylinder ..... lb. per sq. in... | 7.16 | 7.96 |
| 24 | Mean effective pressure, total, reduced to L.P. cylinder, lb. per sq. in... | 21 13 | 21.95 |
| 25 | Mean exhaust pressure L.P. cylinder below atmosphere, lb. per sq. in... | 12.74 | 12.8 |
| 26 | Mean vacuum in condenser below atmosphere, lb. per sq. in... | 13.88 | 13.76 |
| 27 | Revolutions per minute, mean | 61.1 | 63.9 |
| 28 | Piston constant, H.P. cylinder | 4.41 | 4.61 |
| 29 | Piston constant, I.P. cylinder | 10 82 | 11.32 |
| 30 | Piston constant, L.P. cylinder | 30.54 | 31.93 |
| 31 | Indicated horse-power, H.P. cylinder | 205.6 | 221.0 |
| 32 | Indicated horse-power, I.P. cylinder | 221.2 | 231.7 |
| 33 | Indicated horse-power, L.P. cylinder | 218.6 | 255.0 |
| 34 | Indicated horse-power, mean total | 645.4 | 707 7 |
| 35 | Coal burnt per minute ..... lb... | 15.7 | 16.3 |
| 36 | Coal burnt per hour ..... lb... | 942 | 978 |
| 37 | Coal burnt per square foot of actual grate area per hour.. lb... | 22.4 | 23.3 |
| 38 | Coal burnt per square foot of total heating surface per hour ..... lb... | 0.298 | 0.310 |
| 39 | Coal burnt per indicated horse-power per hour ..... lb... | 1.46 | 1.38 |

TABLE II.—COMPARATIVE RESULTS, &c.—Continued.

| — | Trial made by | Research Committee. | Chief Engineer. |
|---|---|---|---|
| 40 | Carbon value of 1 lb. of coal as used...lb... | 1.02 | ..... |
| 41 | Carbon value equivalent per indicated horse-power per hour...lb... | 1.49 | ..... |
| 42 | Feed water per minute...lb... | 143.6 | ..... |
| 43 | Feed water per hour...lb... | 8,616 | ..... |
| 44 | Feed water per square foot of total heating surface per hour...lb... | 2.73 | ..... |
| 45 | Feed water per pound of coal...lb... | 9.15 | ..... |
| 46 | Feed water per pound of coal from and at 212° Fahr...lb... | 10.63 | ..... |
| 47 | Feed water per pound of carbon value from and at 212° Fahr...lb... | 10.42 | ..... |
| 48 | Feed water per indicated horse-power per hour...lb... | 13.35 | ..... |
| 49 | Calorific value of 1 lb. of coal as used...thermal units... | 14,830 | ..... |
| 50 | Percentage of line 49 taken up by feed water... | 69.2 | ..... |
| 51 | Percentage of line 49 carried away by furnace gases... | 16.2 | ..... |
| 52 | Percentage of line 49 lost by imperfect combustion... | 0.0 | ..... |
| 53 | Percentage of line 49 used in evaporating moisture in coal... | 0.0 | ..... |
| 54 | Percentage of line 49 unaccounted for... | 14.6 | ..... |
| 55 | Heat taken up by feed water per minute...thermal units... | 161,100 | ..... |
| 56 | Heat turned into work per minute...thermal units... | 27,590 | 30,250 |
| 57 | Heat taken up by feed water per indicated horse-power per minute...thermal units... | 249.6 | ..... |
| 58 | Efficiency of boiler (line 50)...per cent... | 69.2 | ..... |
| 59 | Efficiency of engine (line 56 ÷ line 55)...per cent... | 17.1 | ..... |
| 60 | Efficiency of engine and boiler combined (line 58 × line 59)...per cent... | 11.8 | ..... |
| 61 | Mean velocity of steam through water surface in boilers per minute...ft... | 1.61 | ..... |
| 62 | Space occupied by boilers per indicated horse-power...cub.ft... | 4.15 | 4.15 |
| 63 | Weight of engines, boilers, &c., with water, per indicated horse-power...tons... | 0.31 | 0.31 |
| 64 | Clearance volume, H.P. cylinder...per cent... | 12.41 | 12.41 |
| 65 | Clearance volume, I.P. cylinder...per cent... | 10.11 | 10.11 |
| 66 | Clearance volume, L.P. cylinder...per cent... | 7.64 | 7.64 |
| 67 | Clearance surface, H.P. cylinder...square feet... | 12.75 | 12.75 |
| 68 | Clearance surface, I.P. cylinder...square feet... | 29.14 | 29.14 |
| 69 | Clearance surface, L.P. cylinder...square feet... | 49.5 | 49.5 |
| 70 | Speed of vessel, mean, during trial...knots... | 8.6 | 9.0 |
| 71 | Mean temperature of outer air...deg. F... | 62 | 91 |
| 72 | Mean temperature of chimney gases...deg. F... | 452 | (672*) |
| 73 | Mean temperature of feed water...deg. F... | 106 | 157 |
| 74 | Mean temperature of boiler steam...deg. F... | 373 | 371 |
| 75 | Mean temperature of circulating water...deg. F... | 55.8 | 78.6 |
| 76 | Mean temperature of discharge water...deg. F... | 75.5 | 94.4 |
| | *Forced Draft Pressures.* | | |
| 77 | In casing near fan...inch of water... | +0.87 | +1.03 |
| 78 | In casing at back of boilers...inch of water... | +0.34 | +0.78 |
| 79 | In ash-pits, mean of 4 gauges...inch of water... | +0.17 | +0.32 |
| 80 | In funnel...inch of water... | —0.25 | ..... |

*Indicated by a pyrometer in uptake of furnace.
J = jacketed. N = not jacketed.

The distance recorded by the registering log during the trial was 135 nautical miles in 16 hours, which is equivalent to a mean speed of 8.4 knots. The distance traversed by the ship during the trial, as measured from the chart, was 138 nautical miles, which is equivalent to a mean speed during the trial of 8.6 knots.

There is added to this report Table II, showing the leading results of the trial of the *Iona* carried out by the committee on July 13 and 14, 1890; and beside these are placed the corresponding figures for the subsequent trial made by the chief engineer on August 15.

Discussion and Comments.—The difference in the temperature of the feed-water on the Committee's trial, and that of the chief engineer, was due to the fact that on the latter the water draining from the jacket and the high-pressure valve chest passed into the feed-pump suction, while on the former it was led into a measuring tank where it cooled down. It was thought that had the feed been equally high in the Committee's trial the coal per I.H.P. would have been decreased to 1.38 lbs. The drain from the jacket, 4.3 per cent. of the total feed-water, was considered by the builder of the engines, Mr. Mudd, very high, heat being given off both to the high-pressure cylinder and the surrounding exhaust passages. The feed-water temperature could be carried as high as 160 deg. to 180 deg. without the pumps refusing duty. If the feed-water had been carried on the Committee's trial as hot as ordinarily, and the coal consumption reduced to 1.38 lb. per I.H.P., this would have raised the boiler efficiency from 69.2 to 73.4 per cent.; and, leaving the engine efficiency at 17.1, this would have raised the total efficiency from 11.8 to 12.55 per cent. The supplementary feed supply on the trial was inordinately great, 6.1 per cent. of the total feed. During a nine days' test made previously to the trial by the chief engineer this amounted to 1.28 per cent. The difference would indicate that during the trial 6,622 lbs. more water was measured than actually reached the boiler, due to leakage of feed-pump gland, &c. This correction applied would make the boiler efficiency 69.8, and raise the engine efficiency from 17.1 to 18, leav-

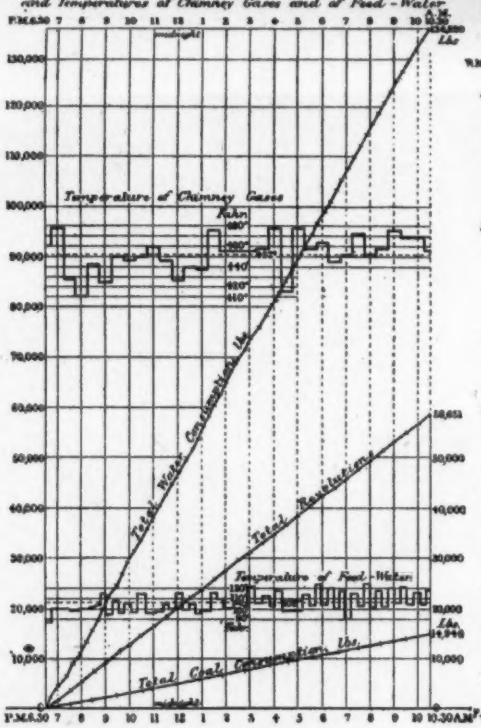
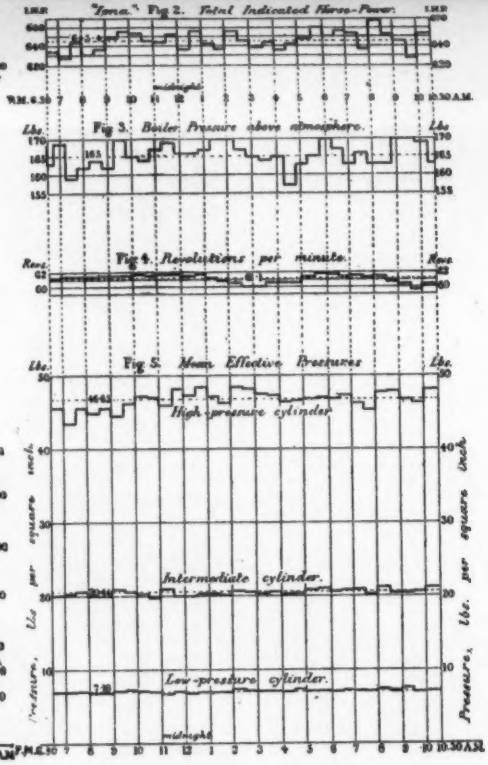
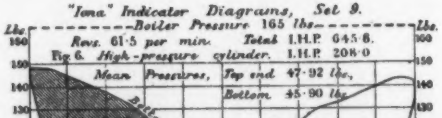
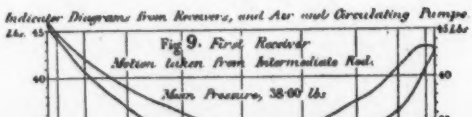


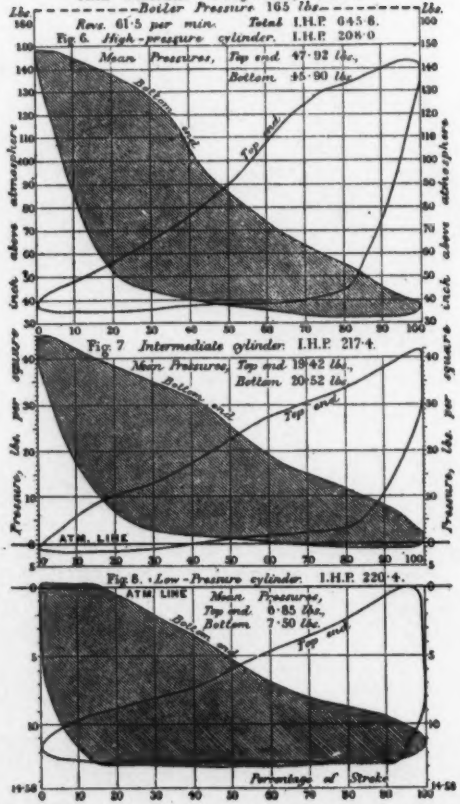
Fig 1. Total Coal, Water, and Revolutions, and Temperatures of Chimney Gases and of Feed-Water
Temperature of Chimney Gases
Total Water Consumption, lbs.
Total Revolutions
Temperature of Feed-Water
Total Coal Consumption, lbs.


"Iona." Fig 2. Total Indicated Horse-Power.
Fig 3. Boiler Pressure above atmosphere.
Fig 4. Revolutions per minute.
Fig 5. Mean Effective Pressures
High-pressure cylinder.
Intermediate cylinder.
Low-pressure cylinder.

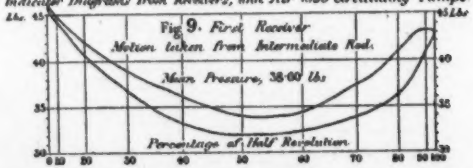

"Iona" Indicator Diagrams, Set 9.
Boiler Pressure 165 lbs
Revs. 61·5 per min. Total I.H.P. 645·6.
Fig 6. High-pressure cylinder. I.H.P. 208·0
Mean Pressures, Top end 47·92 lbs.
Bottom 45·90 lbs.

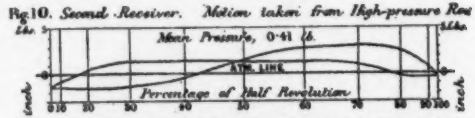

Indicator Diagrams from Receivers, and Air and Circulating Pumps
Fig 9. First Receiver
Motion taken from Intermediate Rod.
Mean Pressure, 38·00 lbs

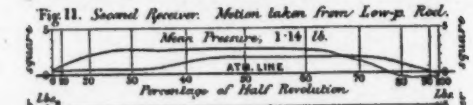
"Jena." Indicator Diagrams, Set 9.

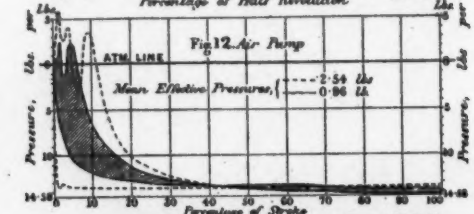


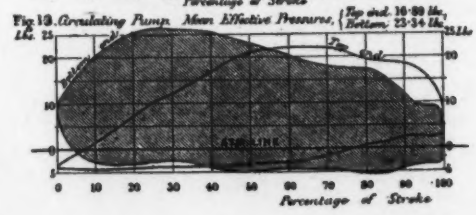

Indicator Diagrams from Receivers, and Air and Circulating Pumps.



ing the total efficiency the same as before, 12.55. Alluding to the percentage of steam and water in the cylinders as given in the report, Mr. Mudd made these figures read, applying the correction to the amount of suplementary feet, which would reduce the water per revolution from 2.35 to 2.23 lbs., as follows:

| | Steam. | Water. |
|---|---|---|
| At 125.4 lbs., . . | 66.8 per cent. | 33.4 per cent. |
| " 19.4 " . | 79.0 " | 21.0 " |
| " 8.6 " . . | 62.4 " | 37.6 " |

Making a correction in these for water drained from the jacket and steam pipe, leaves

| | Steam. | Water. |
|---|---|---|
| At 125.4 lbs., . . | 70.5 per cent. | 29.5 per cent. |
| " 19.4 " . | 83.2 " | 16.8 " |
| " 8.6 " . . | 65.7 " | 34.3 " |

It was pointed out that the boilers were very large for the work they had to do. The ratio of heating to grate surface was 75.1, and this accounted for the low temperature of the funnel gases. These temperatures were found to be 617 in the *Fusi-Yama*, 842 in the *Colchester*, and in the *Tartar*, where also there was provided an extravagant proportion of heating surface, the temperature was 477.

Mr. Willans did not believe that the jacket added much to the efficiency, 4.3 per cent. of the total feed condensed being too small a quantity to have much effect one way or the other. He did not believe in using high-pressure steam to generate low-pressure steam. Better results would have been obtained had all the cylinders been jacketed. In a trial instanced the drain from the jackets amounted to 17½ per cent., and generally the greater the jacket condensation the greater the efficiency of the engine.

In gauging the performance of a boiler, the economy should be considered, and also the rate of steam production. This would give a figure of merit of 29 for the *Iona's* boiler, which is low, that of a locomotive boiler running up as high as 100 on the same basis. It was argued that the draft used could not really be called forced draft. With the damper open as good results could have been obtained with chimney draft. It was shown

that while the *Meteor's* boiler gave off 4.61 I.H.P. to the square foot of water plane, and the *Tartar's* 2.83, the *Iona's* gave off only 2.43. Such a boiler should never prime. Mr. Fothergill stated that he had given up the practice of injecting air through the furnace door in favor of the practice of introducing a jet into the combustion chamber, because in the former case either did the air flow along the furnace crown and keep it cool, or it produced intense local heating. He further argued for short grates as being better suited for burning inferior coal.

The "Engineer" of May 8, after pointing out the fact that, while the indicator cards showed the presence in the high pressure cylinder of about 30 per cent. of water at the beginning of the stroke, and of only about 5.5 per cent. at release, and commenting on the impossibility to account for this, seeing that 18 per cent. should have been condensed to do the work against external resistance and back pressure, while only 4.3 per cent. were condensed in the jacket, goes on to discover a possible connection between initial condensation and economy. The engines of the *Iona* were exceptionally economical, using only 12.69 lbs. of steam per I.H.P., yet the initial condensation amounted to 29.5, according to Mr. Mudd's figures. In the engines of the *Meteor* the initial condensation was only 23 per cent., but the consumption of feed water was 15 lbs. per I.H.P. per hour. In the *Fusi-Yama*, with compound engines, the cylinder condensation with unjacketed cylinders was only 11.8 per cent., but the water used was 21.17 lbs. per horse per hour. In the *Colchester*, with compound engines, cylinders unjacketed, the condensation was 28 per cent. and the feed water 21.73 lbs. The *Tartar* has triple-expansion engines, all cylinders jacketed; the initial condensation was 54 per cent., but the steam consumed amounted to but 16.5 lbs. per I.H.P. The total feed was nearly 20 lbs., but Prof. Kennedy stated that the boiler primed, which statement was contested at the time. The *Tartar* trial is, therefore, to be left out of account as untrustworthy. Enough remains to show, however, that only the most shadowy connection seems to exist between cylinder condensation, the effect of jackets, and the consumption of steam. The "Engineer" concludes with these

words: "It is beginning to be recognized as a truth that condensation and evaporation of vapors depends, to a certain extent, on some specific influence in the shape of an attraction or affinity, existing between certain solids and certain vapors; and far fetched as the notion may appear, there is yet a possibility that some of the remarkable and unaccountable differences in the efficiencies of steam engines, apparently working under identical conditions, are due to some subtle action between the metal of the cylinder and the steam. It is possible, for example, that a very highly polished smooth cylinder may be far less efficient as a condenser than one the working face of which is rough and scored. As bearing on this point, we may direct attention to the very curious and suggestive discovery made by Mr. Bryan Donkin, namely, that a copious supply of oil to a cylinder has precisely the same effect as jacketing in preventing cylinder condensation." We may add, that the experiments made in this country by Prof. Thurston on pickling the steam ports and spaces is a further step in the same direction.

The figures given above were altered in a later issue of the same paper, attention being directed to certain inaccuracies by Mr. Mudd, but the comparison is not affected thereby. The change arose from a modification in the method adopted for obtaining the relative weights of steam and water in the cylinder at any point of the stroke, viz: the feed water measured constitutes a part; the weight compressed and present again during expansion constitutes another, and a certain (assumed) amount of water in the compressed steam constitutes the remainder of the total weight present. This being known and also the pressure at any point, the quantity existing as steam is calculated as usual, giving the amount of water by difference.

In connection with some of the points brought up in the discussion on the *Iona's* trial, some of the results reported in the "Zeitschrift des Vereines Deutscher Ingenieure" of November 9, 1889, *et seq.*, on a test of a single-cylinder condensing Corliss engine, may be of interest. The engine was doing its ordinary every-day work, and the tests extended over many months, each single test lasting at least ten hours. The engine was tried with

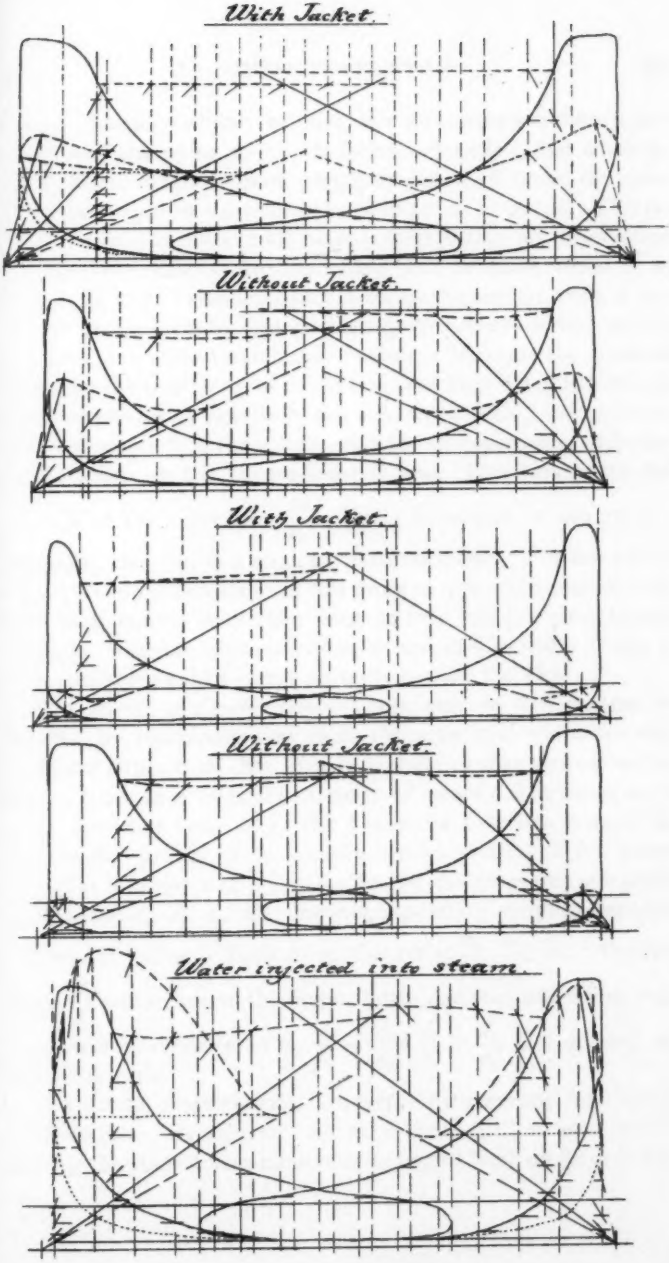
high compression, jacket in use; with high compression and without the jacket; with low compression with jacket and without, and with high compression and jacket in use while the moisture in the steam was measurably increased by injecting water into the steam pipe.

*Effect of Water in the Steam.*—The amount of moisture in the steam has a decided influence on the condensation in the jacket, which increases with the increase of entrained water.

During two trials with water injection (one boiler in use) there were furnished by the boiler per hour 3,506 and 2,850 lbs. respectively. There were injected 311 and 181 lbs. of cold water. This condensed 91 and 53 lbs. more, making the total of added moisture 402 and 234 lbs., which is 10.05 and 7.73 per cent. of the total weight per hour. During these trials the jacket condensation amounted to 1.25 and 1.1 lbs. per I.H.P. per hour.

Trials similar to the above as to pressure, ratio of expansion, &c., gave, with one boiler in use and steam consequently rather wet, a jacket condensation of .77 lb., and with two boilers in use and drier steam, .66 lb. The excess of condensation during the trials with injection of water into the steam over the last-named one is 89.0 and 66.0 per cent., while the excess of condensation with one boiler in use over that when both furnished steam is 16.6 per cent. The influence of the presence of water in the steam is here apparent. It is not, however, so well marked at other times, the difference, under otherwise identical conditions, with one boiler in use and with both being frequently so small as to be covered by the errors of observation.

*Effect of Moisture in the Steam on the Expansion Line.*—The influence shown to be due to moisture in the steam on increase of condensation in the jacket must also become apparent in the expansion line in increasing the pressure ordinates toward the end of the expansion. This is found to be true whenever moisture is increased by shorter cut-off or use of only one boiler. The deviation from a hyperbola passed through the initial point of the expansion curve with the line of no volume (clearance line) as asymptote is, however, very small at all times, amounting in the case of the trials with water injection to only .2 lb. as a maxi-


With Jacket.
Without Jacket.
With Jacket.
Without Jacket.
Water injected into steam.

mum. During ordinary working this difference was more nearly equal to from .05 to .07 lb. It follows, therefore, that when differences are found to exist greatly in excess of these, the cause must be sought in want of tightness of valves or piston. This the author found verified in all cases coming under his observation.

*Expansion Lines.*—An interesting way to study these is by means of their "characteristic" line, as the author calls a line which is obtained by first projecting the points of intersection of the curve with a number of ordinates back on the ordinate marking the beginning of expansion, and then drawing through these points radial lines from the zero point of the clearance line. The intersection of these lines with the corresponding ordinates gives points on the "characteristic" line. This line shows the change of the expression $\frac{pv}{v_1}$. For a hyperbola, where $pv = a$ constant, this line is a straight horizontal line. For the curves of expansion it departs from this more or less. It is noteworthy that with jackets in use the characteristic line is almost always straight, whereas without jacket in use, it frequently drops at first and always rises rather abruptly toward the end.

*The Curves of Compression.*—These may in like manner be studied by examination of their characteristic, which we may compare either with that of a hyperbola or that corresponding to the curve of a constant quantity of steam. The latter starts from the same ordinate as the hyperbola, but rises toward the end of the stroke. It is constructed by plotting a few points found as follows: Let $p_1$, $p_2$, $p_3$, &c., be the pressures at certain points, and $d_1$, $d_2$, $d_3$, &c., the corresponding weights per unit volume of steam. Then form the ratios $\frac{d_1}{p_1}$, $\frac{d_2}{p_2}$, &c.; the first ratio, $\frac{d_1}{p_1}$, will represent the first ordinate, and the succeeding ordinates will be represented by $\frac{d_1}{p_1} \div \frac{d_2}{p_2}$; $\frac{d_1}{p_1} \div \frac{d_3}{p_3}$, &c., plotted on the same scale.

The actual characteristic, however, differs greatly from either of these lines. It starts in tangent to the radial line used in projecting the point where compression begins back on its ordinate,

drops for a while, and then rises rapidly to a maximum, which is reached shortly before the end of the stroke, after which there is an abrupt drop. There is therefore first a condensation, followed by a re-evaporation, due doubtless to the temperature of the metal being higher than that of the steam at what was the admission end of the cylinder. Finally these same metallic walls absorb heat and lead to the final condensation, which would be much lessened were the cylinder head jacketed. The jacket around the cylinder barrel has little influence one way or the other, although its action is still perceptible from the more gradual drop when the final condensation sets in. Although the final pressure of recompression approximates closely that calculated from $pv$ = constant, yet, at the point where the weight of steam apparent is at a maximum, this weight is from 10 to 15 per cent. in excess of that apparent at the beginning of compression, so that the work done on the cushion steam calculated on the basis of $pv$ = constant gives a result too small.

*Economy due to Compression.*—A high degree of compression is desirable to insure the smooth running of an engine, and even should it prove uneconomical, which it may, when it is considered that the work of compression done by the flywheel is a second time subject to the losses by friction, it would still commend itself. The tests gave the following results: With a low degree of compression the weight of steam compressed per hour was from 83.6 to 88 lbs. Subtracting from this the weight represented by the clearance volume filled with steam of a pressure equal to the back pressure leaves from 63.6 to 68.2 lbs. saved. The work done in compressing this steam amounted to about 2.9 I.H.P., a saving of 22 to 24 lbs. per I.H.P. With high compression the weights saved were 211, 249 and 273 lbs. The work expended was 15.2, 17.4 and 21.7 I.H.P., a saving per hour per I.H.P. of 13.8, 14.3 and 12.5 lbs. A point had therefore been reached where the theoretical quantity of steam necessary for the power was increased by the use of compression. Any advantage would then have to be sought in the decrease of initial condensation due to heating the metal in the clearance spaces, and this advantage did exist in a marked degree.

## XXIV.

## THE BURSTING OF THE *CONCORD'S* MAIN STEAM PIPE.

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The recent deplorable accident on the *Concord* whereby two men lost their lives merits very careful investigation, as some points were brought out by it that do not seem to have received the attention that they should. It will be well, therefore, to give first a narrative of the events as they occurred, and then the facts which were developed on investigation, together with such remarks as are suggested by this and similar accidents.

The *Concord* has four main boilers arranged in pairs. They are of the straight through or low cylindrical type, 9 feet 9 inches in diameter, and about 18 feet long without the uptakes. The stop valve connections are on the fronts of the boilers, and as the uptakes are together in the center it puts the valves on the forward boilers some forty feet away in a fore-and-aft direction from the after ones. The steam pipes from the two forward boilers join in front of them, and the main pipe, 8 inches in diameter, then runs along the starboard side of the ship to the after fire-room where it joins one leg of a Y pipe, the other coming from the after boilers, so that the steam from all boilers or from either the forward or after boilers can go the engines. The valve for shutting off the forward boilers is just abaft the after boiler on the starboard side, and there is consequently a length of somewhat over forty feet from the stop valves on the forward boilers to this valve. The annexed drawing shows the main piping, this valve being marked A. Prior to the accident there was no drain pipe anywhere in this pipe forward of the communicating valve, as the pipe is nearly horizontal, and it was supposed that any steam which condensed while this valve was closed would find its way back into the forward boilers.

This description of the piping has been given at length because, as will be seen presently, it is of the greatest importance in a correct understanding of the cause of the accident.

On the 2d of June the *Concord* was at sea for her final trial under the supervision of the Board of Inspection, and in the early forenoon was working under easy steam furnished by the after boilers. From the description of the piping already given it will be understood that the valve at the end of the pipe from the forward boilers was necessarily closed all this time. It was desired to run the engine at full power about noon, so that fires were started in the forward boilers shortly after 8 o'clock, and, as the hydrokineters had been in use for some time previously, steam began to form about 10.30. Shortly after this (about 10.40), and when the pressures were about 10 pounds in each of the forward boilers, the main stop valves were opened so as to have the same pressure in both boilers and avoid the necessity of constantly consulting two steam gauges. Fires were spread in both forward boilers about this same time, but they were not urged nor were the blowers used. About 11.30 the pressure in the forward boilers was the same as in the after ones, and all the boilers were connected through the auxiliary valves and pipes. The pressure was 110 pounds.

At 11.35 the valve in the after end of the pipe from the forward boilers was opened, or rather the man who was to open it began to do so, for the valve was hardly off its seat before an explosion occurred which has been described as sounding like a greatly intensified rip in a piece of cloth. It was not heard all over the ship, however, as an officer who was in the after part of the wardroom says that he did not know that anything had occurred until he heard the orders given in consequence.

It was at once discovered that something had burst in the forward boiler compartment, as it was filled with steam from the athwartship bulkhead between the uptakes as far forward as could be seen. Strangely enough, although the rupture in the pipe was only about 10 feet from the door in this bulkhead, the steam did not come aft. About five minutes after the explosion the body of a man was seen near the door, and it was hauled out

with fire tools, it being impossible for anybody to venture into the forward compartment on account of the intense heat.

As soon as the explosion occurred, the valve which had been started from its seat was closed, and so were the main stop valves on the forward boilers by means of the deck spindles; but the steam continued to pour out, so that they were again opened, as it appeared to make no difference. Cold water was pumped into the after boilers and the safety valves were raised, so as to get rid of the steam as quickly as possible. In about ten minutes after the accident the pressure had fallen to about ten pounds in the after boilers.

It was about two hours and a half, however, before it was possible to get into the forward compartment so as to ascertain what had given way. The examination showed that the pipe from the forward boilers had ripped just forward of the bulkhead between the boiler compartments for a length of about twenty-eight inches right in the brazed seam, and that at the after end of this longitudinal rip there was a circumferential one about seven inches long. This is indicated on the accompanying drawing. Some eight inches from the after end of the seam rip there was a tapped hole, showing that a copper plug had been screwed in.

The water was not visible in the gauge glasses of the forward boilers, but subsequent examination, when the boilers were cold enough to enter, showed that in the starboard boiler the water was about fifteen inches below the top of the combustion chamber, leaving exposed five rows of tubes, and in the port boiler it was about eight inches below, exposing two rows of tubes. The fire side of the upper furnaces showed a red and rusty appearance, but there was no sign of overheating or bulging. To complete this part of the subject, it may be said that the boilers were carefully inspected and tested under hydrostatic pressure at the New York Yard, and there was not even a leaky tube. In explanation it may be said that just before the accident there were about three cocks of water in the forward boilers, or some fifteen inches above the highest heating surface, and that the natural tendency of the steam filling the fire room was to deaden the fire

so that the evaporation was not so fast as under normal conditions. Moreover, during the fall of the water level, while the fires were brightest, each inch of depth corresponded to a much greater volume of water than after the heating surface was reached, so that from all the conditions it seems probable that the water did not get below the highest heating surface for more than an hour after the explosion.

The presence of the tapped hole in the line of brazing of the ruptured pipe at once suggested faulty workmanship, but such inspection as was possible while the pipe was in place did not support this belief, as the metal appeared good and the brazing, except in this one spot, appeared well done.

The Engineer-in-Chief went at once to Norfolk to make a personal inspection, and as soon as he had examined all the parts of the piping and had heard the conditions of raising steam and connecting up the boilers, he pronounced the accident as due to a water ram or water hammer.

A court of inquiry was subsequently appointed, and after exhaustive investigation they gave it as their opinion that the cause was the same as assigned by the Engineer-in-Chief, a water ram.

From discussions in regard to this matter of water rams, it has seemed to the writer that there is not as clear an idea of the *rationale* of these occurrences as should obtain, and he has therefore added entire an excellent article on this subject by Dr. Chas. E. Emery, published some four or five years ago, and, so far as the writer is aware, the only published theory of these phenomena.

From the explanation there given of a water hammer, and from the account which has been given of the piping and the state of affairs, it will be seen that all the conditions for this occurrence existed. There can be little doubt that the burst pipe was more or less filled with water while steam was gradually reaching the pressure in the after boilers and after the main stop valves on the forward boilers had been opened. When the communicating valve was eased off its seat, the current was started, a steam bubble formed and collapsed, and the water, rushing with violence against the side of the pipe, caused its rupture.

In thus stating the conclusion at which the court of inquiry

arrived, it is not meant to imply that their examination was purely one of theories to ascertain the most probable. On the contrary, the section of the injured pipe containing the rip was removed from the ship and subjected to the tests reported as follows by the Board:

"In pursuance of the board's determination to thoroughly examine and test the injured pipe, it was taken to the Navy Yard and the part that remained sound (4 feet 8 inches long) was prepared to be tested by hydraulic pressure. A $\frac{3}{8}$ inch screw plug was screwed into the seam, so that the same initial point of weakness might be furnished that was in the original pipe. Everything being ready on the morning of the 12th inst., the test took place. Pressure was furnished by the accumulator of a riveting machine in the Steam Engineering boiler shop. At 600 pounds the $\frac{3}{8}$ inch plug began to leak. At the same pressure the rivets in the end flange, put in by the contractors, began to leak. At 1,200 pounds one of the $\frac{7}{8}$ inch bolts holding on the $\frac{5}{8}$ inch steel plate ends broke, letting one of the gaskets out. No further application of pressure could be had, and the pipe was examined. The $\frac{5}{8}$ inch steel heads were bulged $\frac{1}{2}$ inch, the ends of pipe, which had been annealed, increased about $\frac{3}{4}$ inch in diameter, and the contractors' flange was cracked and spoiled, but the *brazed seam* did not start.

"For the purposes of this Board this test is sufficient to establish the integrity of the brazing of what was left of the pipe after the disaster.

"It will be seen that the seam which was the suspected part stood uninjured at seven and a half times the working pressure.

"It appeared in the testimony that 300 and 170 pounds had been applied to this pipe in water pressure, and that 168 pounds of steam pressure had been used. After observing this test and weighing the evidence we do not believe that the broken pipe gave out at 110 pounds steam pressure, which, it will be remembered, was the pressure registered on the gauges at the time of the disaster. We are then compelled to look for some other cause. After careful examination of drawings and their verification and the weighing of evidence we arrive at the opinion

that the disaster was caused by water hammering, or, as it is sometimes called, water ramming."

Besides these tests the board made some to determine the quality and strength of the material. In one the test piece was cut so as to bring the brazed seam in the part likely to break, and a piece of the pipe was taken where the brazing looked rather poor. The break occurred just outside the brazing and at a load of about 4,600 pounds per running inch of pipe (say about 21,000 pounds per square inch), corresponding to a pressure of between 1,100 and 1,200 pounds in the pipe. Another test piece was taken from material of the solid part of the pipe, and this broke at about 29,300 pounds per square inch. Both pieces were from the immediate vicinity of the rupture. The pipe was No. 5 B. W. G. thick (.22 inch).

It seems quite clear, as a result of the investigation and test of the damaged pipe, that it was amply strong to stand the regular working pressure, and that in all probability there would never have been any accident if the pipe had been fitted with proper drains. It also emphasizes the importance of looking after the drainage of all pipes where steam may condense and cause such an accident as carefully as we would inspect to discover defective material or workmanship. There seems no particular reason why the rupture might not have occurred in the after part of the pipe, and if it had, instead of two men killed there would probably have been half a dozen, as the engine room as well as the after fire room would have been filled with steam. Perhaps, however, we should be thankful that the lesson was taught at such small expenditure of life and money, as, barring the section of pipe, some lagging and some articles in the magazines which were injured by water, the damage to machinery and hull was very slight.

It is also gratifying to know that the accident was not due to defective workmanship or material, as the ruptured steam pipes on the *Elbe* and the *Lahn* a few years ago seemed to cast considerable suspicion on the safety of brazed joints for thick copper pipes under high pressures.

In this connection it will probably not be out of place to give

some extracts from the excellent paper read before the Institute of Naval Architects by Mr. William Parker, Chief Engineer, Surveyor to Lloyd's Register, on the accidents on these two ships with some of the comments thereon. They will be found following the paper by Dr. Emery, already spoken of.

Another accident to a copper steam pipe, whose bursting cost the lives of seven men, occurred on the English mail steamer *Jumna* in December of last year. It appears that there are three boilers, and on the day of the accident steam had been raised on all of them before they were connected. When the desired pressure was reached, the stop valves were opened, and about a minute and a half afterwards there was a rip (near the seam) in the pipe leading from one of them, which let out an enormous volume of steam and scalded these seven men so badly that they subsequently died.

The rip was in a bend just above the stop valve, and the investigation held by the Board of Trade officials disclosed that for a length of about 6½ inches the metal had a thickness of only .065 to .120 inches, while the designed thickness was .22 inch. It was also found that the general workmanship of the brazed seam in this bend was very poor. The abstract of the report in the papers does not go into the cause of the accident beyond this statement of defective material, but from the sketch of the piping which accompanied the notices it would seem (assuming that there are no drains, and none are indicated,) that the immediate cause of the accident may have been, as on the *Concord*, a water hammer. It seems very likely that no drains were fitted, as the pipe rises all the time after leaving the stop valve, and there would be a very natural disposition to suppose that the water would drain back to the boiler. There were, however, several bends in the pipe. It should be added that the *Jumna* was lying in one of the London docks at the time of the accident.

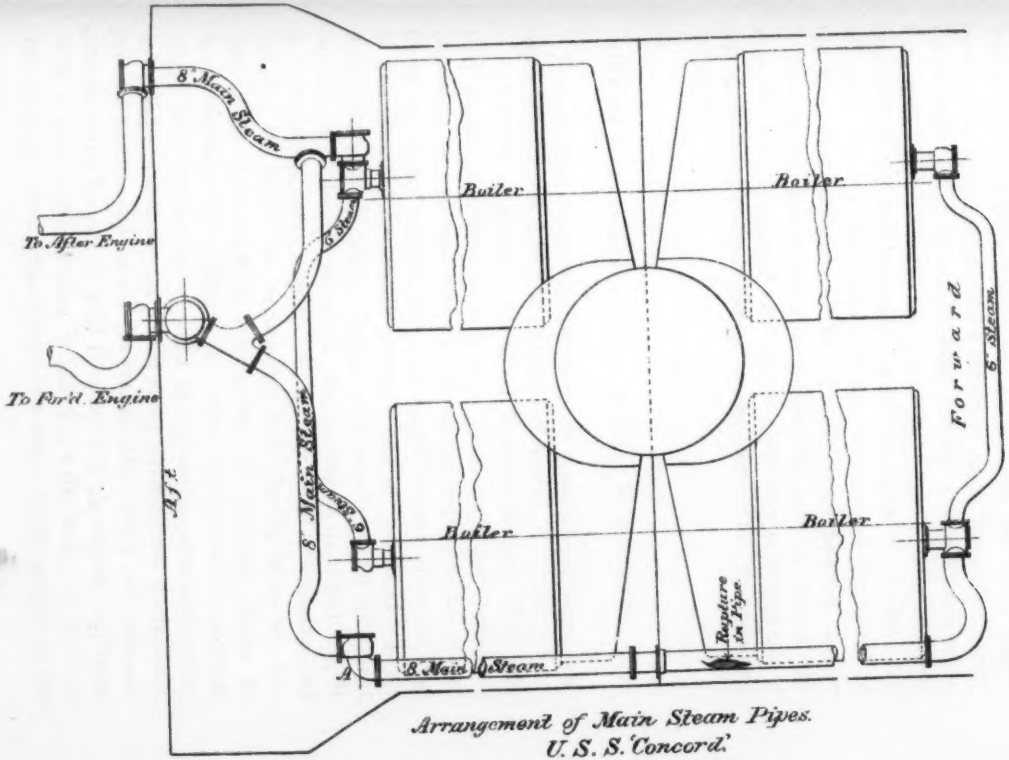
The writer's attention has been called to the fact that several ruptures of steam pipes occurred at different times on the U. S. S. *Tennessee*, which, in the opinion of competent engineers who were then on board, were due to water hammer.

The arrangement of piping in this vessel was very bad and

calculated to cause accident from water hammer. The boilers connected to steam drums at the back, and these drums connected by straight pipes without valves to long pipes which ran through the uptakes, with the idea that the steam would thereby be superheated. There was a double pipe in each uptake (the fire room being fore-and-aft with a set of boilers on each side), the lower one having the connections to the drums and the upper one being connected with the lower one on the opposite side of the fire room by a cross pipe at the end of the fire room away from the engines, while the two upper pipes were joined at the end next the engines by a copper pipe which led to the engine room. There were small drains, perhaps $\frac{3}{4}$ of an inch in diameter, but these were inadequate.

The first accident was a rupture of one of the upper wrought iron pipes in the uptake, and this was unmistakably due to water hammer, because the noise was distinctly heard. It occurred just after steam had been raised and the safety valves had been lowered. The hammering in the pipes was very great and caused apprehension, but as the rupture was in the uptake nobody was injured. This accident occurred at sea while the vessel was on her way to Europe. A second rupture occurred in the copper pipe joining the two upper wrought iron pipes while the vessel was at Gibraltar, and while preparing to leave that place. This time the immediate cause was the opening of a valve to admit steam for warming the engines, although efforts had been made to keep the pipes clear of water by looking carefully after the drains. This second accident made it clear that the drains were not of sufficient size, and in consequence 2-inch drains were put in and great care taken that they were kept open while raising steam. Water would come out of them in great quantities. After they were fitted no further accidents happened to the forward set of boilers.

There were also four boilers abaft the engines with the same arrangement of pipes in the uptake, but these boilers were rarely used because the ship usually went under reduced power and it was more convenient to use the forward boilers. In consequence of their infrequent use, these pipes were not fitted with the larger




*Arrangement of Main Steam Pipes.*
*U. S. S. 'Concord.'*

drains, and it is probable that during the refit after the first cruise the matter was forgotten, as there had been no trouble for so long. In 1883, however, it became necessary on one occasion to use these after boilers, and at that time one of the upper pipes in the uptake ripped for a length of about fifteen feet along the riveted seam. A copper pipe making a direct connection from one of the boilers to the upper pipe in the uptake was also ruptured at the same time, and caused the death of an officer by scalding. It was the opinion of the engineers of the vessel at the time that the accident was due to water hammer, and it is also the opinion of other engineers who have served on the ship and are thoroughly conversant with the arrangement of piping and the circumstances of raising steam when the accident occurred.

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## Water Rams in Steam Pipes; by Chas. E. Emery, Ph. D.

The principal cause of accidents in the operation of large, long steam pipes arises from the presence of water.

If steam be admitted at the top of a vessel partially filled with cold water, condensation will take place until the surface is somewhat heated, and this, in connection with a cloud which forms above the surface, will retard rapid condensation, so that in due time the full steam pressure can be maintained above water cold at the bottom. This phenomenon is not an infrequent occurrence in boilers in which the circulation is defective. It is therefore perfectly safe to heat up any vessel containing cold water, if the steam can be admitted from the top upon the surface of the water and so maintained. If, however, steam be blown in below the surface of the water a bubble will be formed, which will increase in size until its surface becomes sufficiently extended to condense the steam more rapidly than it can enter, when a partial vacuum will be created, the bubble will collapse, and the water flowing in from all sides at high velocity, will meet with a blow forming what is called a water ram. In blowing into a large vessel these explosions occur in the middle of the

mass, and create simply a series of sharp noises. If, however, steam be blown into a large inclined pipe full of water, it will rise by difference of gravity to the top of the pipe, forming a bubble, as previously stated; and, when condensation takes place, the water below the bubble will rush up to fill the vacuum, giving a blow directly against the side of the pipe. As the water still further recedes the bubble will get larger, and move farther and farther up the pipe, the blow each time increasing in intensity, for the reason that the steam has passed a larger mass of water, which is forced forward by the incoming steam to fill the vacuum.

The maximum effect generally takes place at a "dead end," as it is called, or where the end of the pipe is closed. Even if the water does not originally extend to the "dead end," if the pipe near it be once filled with steam which has bubbled through water on its way to that point, there may be sufficient cold metal to condense it, so that the collapse will take place on the same principle as before, and the whole mass of water in the pipe be driven by the incoming current of steam against the end, sometimes with tremendous force, the effect being to cause leaks, and sometimes rupture the pipe or break out the end connections. It is not necessary, either, that the end of the pipe be closed. In fact, under certain conditions, a more forcible blow is struck when the end of the pipe is open, as, for instance, when a pipe crowned upward is filled with water, one end being open and the steam introduced at the other. A bubble will in due time be formed at the top of the crown, when the water will be forced in by atmospheric pressure from one end and by steam pressure from the other, and the meeting of the two columns frequently ruptures the pipe. Evidently, too, the same action can occur without difficulty in a level pipe, but, as previously stated, cannot in a pipe which descends away from the entering steam, so that the latter is *always* above the water.

It is evident from the above that it is always desirable in turning steam on an inclined plane to introduce it from the top, and let the water out of the bottom of the slope. When this can be done, any workman can be trusted to attend to it. Frequently,

however, there are undulations in the pipe, and at times mains which may contain water have to be heated by letting the steam in at the lower end, and letting the water out before the steam is admitted. The same thing can be done with underground pipes, and provisions for this should always form a part of the plans when it is known that a pipe will have to be heated up in this way.

In practice, however, a street system contains so many absolutely necessary details that a provision of this kind will not be originally provided for, and at times it will occur that a main which it was expected to heat from the top of the slope, may, from something being out of order, necessarily be heated from the other direction. Difficulties also occur in small pipes where the extra labor and expense required to provide special drains for overcoming this difficulty would not be warranted, particularly as another solution of the difficulty is available, even with pipes of considerable size.

If a blow-off opening be provided at one end of a main to be filled with steam, even if such blow-off be at the higher end, any water in the main can be driven out of the blow pipe, provided the steam valve be opened sufficiently wide to keep the pressure continuously maintained against the water. The explanation of this is that if the steam supply be limited, the water will run back under portions of the steam, forming bubbles which may suddenly collapse and produce water rams; but if the steam supply be practically unlimited, or at least sufficient, the steam will force the column of water back along the bottom of the pipe, as any vacuum formed will be filled by the steam driving back the water. There will be a series of small explosions, which will scarcely be heard and do no harm, and the seething wall of water will be continually forced forward and finally out of the pipe.

Note the distinction in the two methods of operation necessary to suit the conditions. When the steam is on the top of the water, it may be turned on as slowly as desired, and it is better to turn it on slowly, as thereby the heavy castings are heated slowly, and are not so liable to be strained. But when steam *must* be turned in the lower end of a descending

pipe which *may* be filled with water the valve must be opened sufficiently to establish a definite current and keep up the pressure. This will not require the valve to be wide open, but the result will be substantially as though it were open. Practical engineers, who on sea and land have had to do with turning on steam in pipes, naturally recoil from turning steam quickly into any pipe, and it is very hard to explain to them the difference. I have had to take a party of men of this kind, state the reasons for action, and in one case I recollect using as an illustration, that if a farmer with a pitchfork could get an officer on the run, the latter could not draw his sword, turn, and defend himself, as he would be run through before he came to close quarters. The principle applies to the water in an ascending pipe. The column of water once started, the steam, if the supply be made sufficient, follows it up so closely and in such volume that no condensation can take place sufficiently to stop the onward movement.

The clearing of a pipe in this way requires nerve and judgment, but I have seen considerable cold water driven up hill out of a six-inch pipe 1,400 feet long with a difference of elevation at the two ends of fully twenty feet, by letting steam in at the lower end and blowing the water out on the street through a two-inch blow-off pipe. The blow-off pipes are made no larger than this, even for mains fifteen and sixteen inches in diameter, but I do not consider that it would be safe to attempt to clear an ascending main of this size with this size of blow-off pipe. All these mains are more nearly level, have blow-offs at low points, near the valves, and can be blown off by putting steam in at or near the summit. In heating up an eleven-inch pipe, only 400 or 500 feet long, from the bottom, I have had the flange taken off the extreme end in order to give the water free exit and prevent the possibility of a ram.

## Extracts from Paper by Wm. Parker, Esq., on the Bursting of the Copper Steam Pipes of the Steamers *Elbe* and *Lahn*.

The S. S. *Elbe* was built in 1870 by Messrs. J. Elder & Co., and fitted with ordinary compound engines. In February of

1887 she was placed in the hands of Messrs. Oswald, Mordaunt & Co., of Southampton, for the purpose of being fitted with new boilers to work at a pressure of 150 pounds per square inch, and of having her compound engines converted into triple-expansive ones. This work was completed and the vessel was running her official trial in Stokes Bay, when the main steam pipe abreast of the after boilers burst, and all in the stoke hole at the time, numbering eleven persons, were killed by the sudden outrush of steam.

The S. S. *Lahn* is a new vessel built by the Fairfield Shipbuilding and Engineering Company in 1887. In March of that year, while on her second voyage across the Atlantic, her steam pipe burst in a somewhat similar manner, but the loss of life was less in this instance than in the case of the *Elbe*.

It was natural to suppose that the bursting of copper steam pipes, such as those of the *Elbe* and the *Lahn* would be found attributable to defective workmanship or material, and that due care had not been exercised in the manufacture of the pipes; but these investigations go to show that elements of very serious danger enter into the ordinary methods of making brazed copper pipes, especially when intended for high pressure steam.

The steam pipe which exploded on board the *Elbe* was $9\frac{3}{8}$ inches diameter inside and 6 feet 6 inches long. The thickness of the copper was .276 of an inch, corresponding to No. 2 Imperial wire gauge. It was brazed in the usual manner, with a lapped joint, and to all appearance seemed a well made pipe. The copper had been obtained from a first class manufacturer in Birmingham, and analysis has shown it to be chemically of the best quality.

The pipe itself had been tested by hydraulic pressure on two occasions, once for the satisfaction of the builders to a pressure of 300 pounds per square inch, and a second time by the owner's representative to 350 pounds per square inch. These tests had been sustained in a perfectly satisfactory manner, and so far as could be judged, every care had been exercised by the makers to produce as good a piece of wormanship as possible.

Test pieces were cut from the exploded pipe, and the copper

away from the locality of the brazing was found to have an ultimate tenacity of 33,000 pounds per square inch, with an elongation before fracture of 33 per cent. in a length of 5 inches, so that the bursting pressure for this pipe, in its cold state, should have been about 1,940 pounds per square inch, being thirteen times the working pressure, or, taking the actual thickness of the copper at the fracture as measured after the explosion, and which was found to be $\frac{3}{16}$ of an inch, the pipe should still have borne an ultimate pressure of 1,220 pounds per square inch, or 8.8 times the working pressure.

In order to ascertain the actual strength of the portions of the pipe adjacent to the exploded part, a piece of pipe, about 30 inches long, was cut from the portion still intact of the length which exploded, and two other similar pieces were prepared from the next adjoining length of the pipe. These short pieces were fitted with flanges, and tested to destruction by hydraulic pressure, and in each case the pipe burst exactly in the same way, namely, through the copper near and parallel to the brazed seam, commencing near to one of the flanges.

The pieces taken from the length where the explosion had occurred gave way under a hydraulic pressure of 780 pounds per square inch, the fracture exhibiting a granular and in part a discolored appearance.

Of the two pieces cut from the next adjoining length of the pipe, one burst at 600 pounds per square inch with the same granular and partly discolored fracture, as if the metal had been injured or partially cracked through during the operation of brazing; and the other burst at 1,140 pounds per square inch, the fracture being granular but not discolored.

This great diversity in the bursting pressures for pipes of the same dimensions and material, and the similarity in the character and position of the fracture in each case, together with the fact that the pressure at which the strongest of these lengths burst was still not more than five-eights of the calculated bursting pressure, seemed clearly to indicate that the material had been injured in the neighborhood of the seam by the operation of brazing.

Some very careful experiments to ascertain the effect of increased temperatures upon sheet copper were made by the Franklin Institute in America as long ago as 1837, when strips of copper were pulled asunder in a properly designed machine at temperatures ranging from 122 degrees to 1,332 degrees Fahr., and, as will be seen by the following table, the falling off in strength as the temperature increased was very considerable, until at 1,332° F., or a bright red heat, the tenacity was *nil*:

*Experiments made at the Franklin Institute, America, in connection with an investigation into the strength of boilers.*

| Temperature above 32 degrees. | Diminution of strength. | Temperature above 32 degrees | Diminution of strength. |
|---|---|---|---|
| 90° | 0.0175 | 660° | 0.3425 |
| 180° | 0.0540 | 769° | 0.4389 |
| 270° | 0.0926 | 812° | 0.4944 |
| 360° | 0.1513 | 880° | 0.5581 |
| 450° | 0.2046 | 984° | 0.6691 |
| 460° | 0.2133 | 1,000° | 0.6741 |
| 513° | 0.2446 | 1,200° | 0.8861 |
| 529° | 0.2558 | 1,300° | 1.0000 |

These results were substantially corroborated by a series of experiments made by Dr. Kirk and myself after the *Elbe* explosion.

From the table it will be seen that at 360° Fahr., or the temperature of steam of 150 pounds pressure, copper has about 15 per cent. less tensile strength than when cold. But allowing for this falling off in tenacity the steam pipe in the *Elbe* at 150 pounds steam pressure should still have had a factor of safety of 8½.

Carefully examined through a microscope, the difference in the structure of the copper in the neighborhood of the brazing where the exploded pipe and those afterwards tested gave way and the structure in that part of the pipe away from the brazing was very marked, and was clearly to be ascribed to the heating of the copper during brazing.

Test pieces were cut from the pipe near to the brazed seam where the rent had occurred, and it was found that whereas the copper from other parts of the pipe showed a tenacity of 33,000 pounds per square inch, with an elongation of 33 per cent. in 5 inches and 59 per cent. contraction of area as recorded above, in these pieces the tenacity was only 24,418 pounds per square inch, the elongation only 4.6 per cent. and the contraction of area at the fracture only 13 per cent. This result was amply confirmed by testing other strips cut from the same locality; they each showed that the copper near the brazing had lost its ductile qualities and much of its tenacity.

In order to obtain further information on this point the following experiments were made:

1. A strip of good copper was cut from a sheet and bent and broken cold.

2. A similar piece was raised to a heat above that necessary for brazing, when it became red short, and broke with its own weight.

3. A piece was raised to the same bright heat as No. 2, allowed to cool, and then broken cold.

4. A piece was raised to the above brittle heat, at which it was partially broken through, then allowed to cool, and the fracture completed when cold.

The appearances of these fractures were very interesting. The first had the fibrous, silky appearance of a good copper; in the second piece the fracture was coarse and blackened by the heat; but the third specimen seemed to have almost completely regained its tenacity and ductility on cooling, the copper in its normal state having a tenacity of 35,212 pounds per square inch, with an elongation of 40 per cent. in 5 inches, and a contraction of area at the fracture of 39.9 per cent., while the third specimen, although it had been raised to a blistering heat, was, when cold, found to have a tenacity of 31,337 pounds per square inch, with an elongation and contraction of area at fracture practically the same as the normal copper, which would appear to show that, although the copper should be overheated in brazing, if not

otherwise injured, its qualities will be substantially regained on cooling.

The fourth piece was partially broken while hot, and the appearance of that part of the fracture was discolored by the action of the heat, but that part that was allowed to cool had its ductility restored, and afterwards broke with a bright fracture.

This experiment was repeated a number of times, with similar results, the fractures corresponding *exactly* in appearance not only with that of the exploded pipe, but also with those of the *other* pipes which were experimented upon, and which burst at pressures of 600 and 780 pounds per square inch respectively.

The only difference in these fractures was the depth to which the partial crack, as shown by discoloration, had extended.

In this manner the exact appearance of the fractures of the exploded pipe, and of those experimental pieces which burst at the low pressures, was *reproduced artificially* by burning the copper and treating it in the way described.

From this it will be seen that, should a copper pipe be overheated during the brazing operation, seeing that the metal becomes perfectly brittle at not much above the brazing heat, the pipe might accidentally be cracked when in this brittle condition; and, although the section of metal still remaining intact might be sufficient to sustain the cold-water test pressure, yet the hot steam and accompanying strains might develop and deepen the crack, and the pipe ultimately give way at the working pressure.

This I consider to be the true explanation of the explosion of the *Elbe's* steam pipe, and also that of the S. S. *Lahn*.

A serious element of danger is thus known to exist in the present practice of brazing large, heavy copper pipes intended to be subjected to the high pressures now so common. It is generally admitted that welds or brazed joints in any material must possess certain elements of uncertainty, and the above experiments show this uncertainty to be greatly increased in the case of copper worked over a fire.

How to eliminate these elements of danger becomes an important question. Various alternatives suggested themselves,

as, for instance, that the copper pipes might be served with wire or have strengthening bands fitted at short intervals (both of which systems I find are now resorted to), or that steel pipe might be used, the seam being riveted.

DR. A. C. KIRK, LL. D.—I will only say one or two words with reference to the brazed copper pipes that we have been for ages in the habit of using. Do not let us forget that they have been very good servants hitherto in times past, and are not to be altogether thrown away. The experiments that Mr. Sinclair made, and which are given in detail in a paper which was read before the Institution of Engineers and Shipbuilders here, show that the ordinary brazed copper pipe made by a good tradesman is a very satisfactory and reliable thing indeed. Further, seeing that out of the large number of high-pressure engines which have been made with brazed steam pipes we have had only two notorious cases of steam-pipe explosion, I think we must admit that their usefulness is not by any means exhausted, and the only thing you can say against them is that their goodness depends much on the skill of the tradesman who does the brazing. You may find within the length of a 12-inch pipe a difference in strength, as Mr. Parker puts it, of perhaps as much as 30 per cent. This has long been known, and a very large margin of strength has been habitually in use. But still with all that, as I say, it is a very good servant. We know that the thickness and the strength of copper pipes in use have not been arrived at by abstract calculation, but have been the result of very long experience, and in that way our present factors of safety for copper pipes cover all the irregularities that are likely to occur when a good tradesman does the work, although the copper is soft; and all I have got to say is that when we substitute unbrazed pipes, whether solid-drawn or whether electro-deposited pipes, we ought to make a reduction in thickness, while at the same time we will eliminate many risks from bad workmanship. From the experiments which Mr. Sinclair made, I should put the proper reduction in thickness roundly at 25 per cent. when the unbrazed pipe replaces the brazed one. I do not propose getting rid of brazing altogether. We have questions of flanges to

deal with, and when we come to large pipes we have the question of bending to take account of. The bent pipe may be less trustworthy than the brazed. I am not aware that there is any information in this world about the bending of a 12-inch copper pipe, and as to how much it is worth after it is bent, or even how the bending is to be done. In that case I presume we would have recourse to brazed copper or brass castings for bends, and content ourselves with straight pipes when solid. It is quite evident that such things as 12-inch solid-drawn pipes can only be applied to a limited degree. Where the ship is made to give way a little to the machinery, main steam pipes may be made straight, with brazed or cast bends; but there are many pipes of large size connecting parts of the engines that cannot be made so, they must be made bent, and in that case they must still be built and brazed. However, the advance which is making in solid-drawn pipes, and no doubt also in the pipes which Mr. Parker has described, will be a very useful advance indeed; but we are not to suppose, as far as we have gone at present, that they will certainly replace brazed pipes.

MR. F. C. MARSHALL.—I do not wish to occupy the time of the meeting further than to confirm to some extent the results described by Mr. Parker in his very able paper. I have sent from Newcastle a piece of an expansion joint, showing precisely the effect that Mr. Parker has described round a considerable portion of its circumference, though not entirely. In order to put the matter precisely, I have requested Mr. Milton, a member of our council, to give me the history of the pipe upon which he has experimented, and with your permission I will read the letter he has sent: "I have sent by passenger train the piece of the copper expansion joint, and also the test pieces cut from it. The history of the piece is as follows: It was delivered here as a circle, and was hollowed cold by a mallet. Then it was heated to a blood-red color, and dished hot with a mallet. A small circle was then cut out of it and the inner edge drawn square, still being worked at blood-colored heat. The flange was then put on it and it was brazed over a flat fire, the upper part having an iron plate cramped on it to keep the heat down and to stiffen

the copper piece to hold it by, the iron plate being clayed over on the inside. The flange was successfully brazed on, and on cleaning the piece afterwards the minute cracks were observed for about five-sixths of the circumference. After cutting the flange off, it has been bent about to observe the effect of the cracks," which are very perceptible and are distinctly described by Mr. Parker. They show that the effect of the heating has been to crack the body of the copper nearly half through, leaving a distinct black crystalline mark where it was opened out. "This history has been gotten from the foreman. The workman who did it is one of his most skilful men. I have had two pieces cut off the expansion joint at that part where there were no cracks, and have had them treated over a fire. The first one, you will see, was on the point of melting, the under side being fused and just ready to drop. On the upper side a piece of spelter (solder) was melted long before the temperature was too much raised. This piece had the overhanging weight of itself when on the point of fusing, but no cracks were developed, and it has been flattened out since without any defects showing themselves. The other piece was, unfortunately, heated just too much and a piece dropped off it, not being able to bear its own weight. This second piece has developed the small cracks similar to those in the original expansion-joint piece. From the flat portion, near the defects, I have had two test pieces cut and tested. One was well hammered, marked *H;* the other annealed, marked *A.* The test results show them to be very good; annealed, 13.9 tons per square inch, with 45 per cent. elongation in 4 inches; hammered, 14.95 tons, with 29 per cent. elongation. I think the heating experiment shows that these defects are only produced at a temperature varying very little from the fusing point, and that the margin of temperature at which the copper is brittle is so small that it is very improbable that it can be reached, except very locally, without some part showing by the blistered appearance that the copper has been overheated." This fully confirms Mr. Parker's view that at certain points—and it is also confirmed by Dr. Kirk—at certain temperatures copper does become disintegrated and produces this effect. I quite confirm Dr. Kirk's

view with reference to the brazed pipes. I should be very sorry to give up brazed pipes after such long service. It is to be observed in the case of the *Elbe*, if I remember rightly, the bursting was not at the soldered joint, but immediately adjoining it, showing that the copper heated.

If the temperature is kept down and the workmanship is properly carried out the brazed pipe is an excellent servant, as Dr. Kirk has said. It is further to be observed that in the *Elbe* pipe the explosion took place near the brazing of the flange; it therefore requires that we should give greater attention to that part of our work; and while it would not be well to dispense altogether with the brazing of flanges, and instead of that to substitute the old procedure again of riveting them together and depending solely on the riveting, I think that it is a question well worth our consideration, and it might be a useful outcome to Mr. Parker's paper. I am sure that we are very much indebted to Mr. Parker for having brought this matter before us. It will induce greater caution on the part of marine engineers with regard to a matter which perhaps has not up to the time of our using very high pressures of steam received the proper amount of attention.

Mr. W. Parker.—When writing and reading this paper I did not think there would be any discussion upon it. My object was to give to the members of this institution the results of the investigations I had made into the causes of the explosions that occurred on board the steamers *Elbe* and *Lahn*, and also to give to the meeting the experience I have gained while investigating the manufacture of electro-deposited copper and comparing its properties with the properties of solid drawn and plate copper, with a view of having pipes in future made without a brazed joint, and thus try to eliminate the element of danger which, to my mind, was undoubtedly the cause of these unfortunate accidents. I am glad to find, from Mr. Marshall's remarks, that he has taken hold of the point I refer to. It is simply this: When a copper pipe is being brazed it is raised to a temperature at which spelter melts, and, comparatively speaking, at a few degrees above this temperature copper becomes absolutely rotten

and will break even with its own weight. It then depends upon the skill of the workman who brazes the pipe whether you have a sound pipe or a pipe having a latent defect in it. If you look at the fractures of these pieces of copper, which fractures have been produced artificially, it will be seen by the discolored portions that the copper had cracked almost through the plate while it was at this temperature and had been holding only by the small portion having a bright appearance. These fractures of specimens of copper correspond exactly with the fractures that appeared not only in the *Elbe's* pipe but also in the other pipes, which while being tested gave way at such alarmingly low pressures. I think you will agree with me that there is very great risk at ending the brazing of copper pipes not at the brazing, but, as Dr. Kirk has put it, outside the brazing. This investigation has shown that latent cracks, such as shown by these specimens, did exist in the pipes that gave way, and that we are always more or less liable to such cracks if we continue to adopt the present practice. Mr. Marshall said, strange to say, these pipes generally give way near the flange. I think that is easily explained. In brazing the seam of a pipe it is done in short lengths; but when brazing a flange the pipe is placed on end and on the fire, and for a short length the pipe is heated all over. This part is made much softer than the other parts of the pipe, and it gives way there first.

| Diameter of Pipe. (inside) | Diameter of Flange. | Thickness of Flange "a" | Bore of Flange. | Thickness of Pipe. | Depth of Boss "b" | Diameter of Boss "c" | Number and Diameter of Holes | | Diameter of Bolt Circle |
|---|---|---|---|---|---|---|---|---|---|
| 1" | 3½" | ¼" | 1⅜" | ¼" | ¼" | 2⅜" | 4 | ⅜" | 3 1/16" |
| 1½" | 6" | ¼" | 2¼" | ¼" | ¼" | 2⅝" | 6 | ⅜" | 4½" |
| 2" | 6½" | ¼" | 2½" | ¼" | ¼" | 3⅛" | 6 | ⅜" | 4 15/16" |
| 2½" | 7½" | 5/16" | 3 1/16" | 5/16" | ¼" | 3 5/16" | 6 | ⅜" | 5⅜" |
| 3" | 7½" | 5/16" | 3 7/16" | 5/16" | 11" | 4⅛" | 8 | ⅜" | 5 13/16" |
| 4" | 8½" | ⅜" | 4⅜" | ⅜" | 11" | 4⅞" | 8 | ⅜" | 6½" |
| 4½" | 9" | ⅜" | 4⅞" | ⅜" | 11" | 5⅜" | 8 | ⅜" | 7" |
| 5" | 9½" | 7/16" | 5⅜" | ⅜" | 11" | 5⅝" | 10 | ⅜" | 7 9/16" |
| 5½" | 10" | 7/16" | 5⅞" | ⅜" | 1⅛" | 6⅛" | 10 | ⅜" | 8⅛" |
| 6" | 10½" | ½" | 6⅜" | ⅜" | 1⅛" | 7" | 10 | ⅜" | 8⅝" |
| 6½" | 11½" | ½" | 6 15/16" | 7/16" | 1⅛" | 7⅜" | 12 | ⅜" | 9⅛" |
| 7" | 11½" | ½" | 7⅜" | 7/16" | 1⅛" | 8⅛" | 12 | ⅜" | 9 11/16" |
| 7½" | 12½" | 9/16" | 7 15/16" | 9/16" | 2" | 8⅝" | 12 | ½" | 10⅛" |
| 8" | 13" | 9/16" | 8 7/16" | 9/16" | 2" | 9⅛" | 12 | ½" | 11" |
| 8½" | 13½" | 9/16" | 9" | ½" | 2" | 9 11/16" | 14 | ½" | 11 5/16" |
| 9" | 14½" | ⅝" | 9½" | ½" | 2" | 10⅛" | 14 | ½" | 12⅛" |
| 9½" | 14½" | ⅝" | 10 5/16" | 11/16" | 2⅛" | 10⅝" | 14 | ½" | 12⅝" |
| 10" | 15½" | ⅝" | 10⅞" | ⅝" | 2⅛" | 11⅛" | 14 | ½" | 13⅛" |
| 10½" | 15½" | ⅝" | 11⅜" | ⅝" | 2⅛" | 11⅝" | 15 | ½" | 13⅝" |
| 11" | 16½" | ⅝" | 11⅞" | ⅝" | 2⅛" | 12⅛" | 15 | ½" | 14⅛" |
| 11½" | 17" | 1" | 12⅜" | ⅝" | 2⅛" | 12 5/16" | 16 | ½" | 14 5/8" |
| 12" | 17½" | 1" | 12 15/16" | 11/16" | 2⅛" | 13⅛" | 16 | ½" | 15⅛" |
| 13" | 19½" | 1⅛" | 13 15/16" | 11/16" | 2⅛" | 14⅛" | 18 | ½" | 16⅛" |
| 14" | 19½" | 1⅛" | 14 15/16" | ¾" | 2⅛" | 15⅛" | 19 | ½" | 17⅛" |
| 15" | 21" | 1⅛" | 15 15/16" | ¾" | 2⅜" | 16⅛" | 20 | ⅝" | 18 5/8" |
| 16" | 22½" | 1⅜" | 16 15/16" | 13/16" | 2⅜" | 17⅛" | 20 | ⅝" | 19⅛" |
| 17" | 23½" | 1⅜" | 17 15/16" | ¾" | 2⅜" | 18⅛" | 21 | ⅝" | 21" |
| 18" | 24½" | 1⅜" | 18⅞" | ¾" | 2⅜" | 20" | 22 | ⅝" | 22½" |
| 19" | 25½" | 1⅜" | 19 15/16" | ¾" | 2⅛" | 21" | 23 | ⅝" | 23½" |
| 20" | 26½" | 1⅜" | 20 15/16" | ¾" | 2⅛" | 22" | 24 | ⅝" | 24½" |
| 21" | 28" | 1½" | 21 15/16" | ¾" | 2⅛" | 23⅛" | 25 | ⅝" | 25½" |
| 22" | 29½" | 1½" | 23" | 1" | 2⅛" | 24⅛" | 26 | ⅝" | 26⅝" |
| 23" | 30½" | 1½" | 24 15/16" | 13/16" | 3" | 25½" | 27 | 1" | 27⅝" |
| 24" | 31½" | 1⅝" | 25½" | ¾" | 3" | 26⅛" | 28 | 1" | 28⅝" |
| 25" | 32⅝" | 1⅝" | 26⅝" | ¾" | 3" | 27⅛" | 29 | 1" | 29⅝" |
| 26" | 33⅞" | 1 11/16" | 27½" | ¾" | 3" | 28⅛" | 30 | 1" | 30⅞" |
| 27" | 35" | 1⅝" | 28⅝" | ¾" | 3⅛" | 29⅛" | 31 | 1" | 31 13/16" |
| 28" | 36½" | 1⅝" | 29⅝" | ¾" | 3⅛" | 30⅛" | 32 | 1" | 32 13/16" |
| 29" | 37⅞" | 1⅞" | 30⅝" | ¾" | 3⅛" | 31 1/16" | 33 | 1" | 34⅛" |
| 30" | 38½" | 1⅞" | 31½" | ⅞" | 3⅛" | 32⅛" | 33 | 1" | 35⅛" |

Note: Holes in flanges to be bored ⅛" larger than di

All flanges to be grooved.

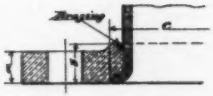
This Table to be used for Pressures of from

| Number and Diameter of Holes | | Diameter of Pitch Circle | Pitch of Holes. | Diameter and Pitch of Rivets. | | Number of Rivets. |
|---|---|---|---|---|---|---|
| 4 | ⅜" | 3 13/16" | 2.99" | | | |
| 6 | ⅜" | 4 ½" | 2.26" | | | |
| 6 | ⅜" | 4 11/16" | 2.42" | | | |
| 6 | ⅜" | 5 1/16" | 2.85" | | | |
| 8 | ⅜" | 5 ⅜" | 2.33" | | | |
| 8 | ⅜" | 6 ¼" | 2.53" | | | |
| 8 | ⅜" | 7" | 2.75" | | | |
| 10 | ⅜" | 7 ½" | 2.36" | | | |
| 10 | ⅜" | 8 ⅛" | 2.53" | ¼" | 1" | 20 |
| 10 | ⅜" | 8 ¾" | 2.69" | ¼" | 1" | 22 |
| 12 | ⅜" | 9 ⅛" | 2.46" | ¼" | 1" | 24 |
| 12 | ⅜" | 9 11/16" | 2.54" | ¼" | 1" | 26 |
| 12 | ⅜" | 10 ¼" | 2.73" | ¼" | 1" | 27 |
| 12 | ½" | 11" | 2.86" | ¼" | 1" | 29 |
| 14 | ½" | 11 ⅛" | 2.39" | ¼" | 1" | 30 |
| 14 | ½" | 12 ⅛" | 2.71" | ¼" | 1" | 32 |
| 14 | ½" | 12 ¾" | 2.85" | 5/16" | 1¼" | 24 |
| 14 | ½" | 13 ⅜" | 2.94" | 5/16" | 1¼" | 28 |
| 15 | ½" | 13 ¾" | 2.88" | 5/16" | 1¼" | 30 |
| 15 | ½" | 14 ¼" | 2.98" | 5/16" | 1¼" | 32 |
| 16 | ½" | 14 ⅝" | 2.90" | 5/16" | 1¼" | 33 |
| 16 | ½" | 15 ⅛" | 3.01" | 5/16" | 1¼" | 34 |
| 18 | ½" | 16 ⅛" | 2.94" | 5/16" | 1¼" | 36 |
| 19 | ½" | 17 ⅛" | 2.98" | 5/16" | 1¼" | 39 |
| 20 | ⅝" | 18 ⅞" | 2.95" | 5/16" | 1¼" | 42 |
| 20 | ½" | 19 ⅛" | 3.12" | 5/16" | 1¼" | 43 |
| 21 | ½" | 21" | 3.14" | 5/16" | 1¼" | 48 |
| 22 | ⅝" | 22 ⅛" | 3.16" | ⅜" | 1½" | 42 |
| 23 | ⅝" | 23 ⅛" | 3.16" | ⅜" | 1½" | 44 |
| 24 | ⅝" | 24 ⅛" | 3.16" | ⅜" | 1½" | 46 |
| 25 | ⅝" | 25 ⅛" | 3.18" | ⅜" | 1½" | 48 |
| 26 | ⅝" | 26 ⅛" | 3.18" | ⅜" | 1½" | 50 |
| 27 | 1" | 27 ⅛" | 3.21" | ⅜" | 1½" | 53 |
| 28 | 1" | 28 ⅛" | 3.22" | ⅜" | 1½" | 55 |
| 29 | 1" | 29 ⅛" | 3.22" | ⅜" | 1½" | 57 |
| 30 | 1" | 30 ⅛" | 3.23" | ⅜" | 1½" | 60 |
| 31 | 1" | 31 ⅛" | 3.24" | ⅜" | 1½" | 62 |
| 32 | 1" | 32 ⅛" | 3.23" | ⅜" | 1½" | 64 |
| 33 | 1" | 33 ⅛" | 3.24" | ⅜" | 1½" | 66 |
| 33 | 1" | 33 ⅛" | 2.87" | ⅜" | 1½" | 69 |

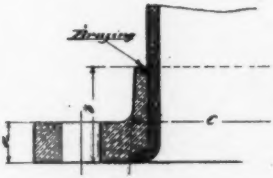
larger than diam. of holes.

grooved.

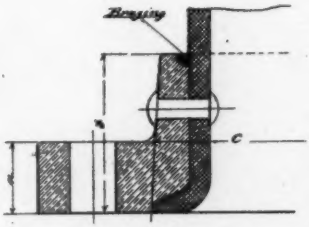
s of from 100 to 160 lbs.




For Pipes of 3" diam. and under




For Pipes from 3½ to 5" inclusive.




For Pipes from 6" to 30" diam. inclusive.

Taper of bores ⅛" to the foot for all pipes above 5" diam. Pipes to be flanged out to suit the contour line of boss continued, as shown.

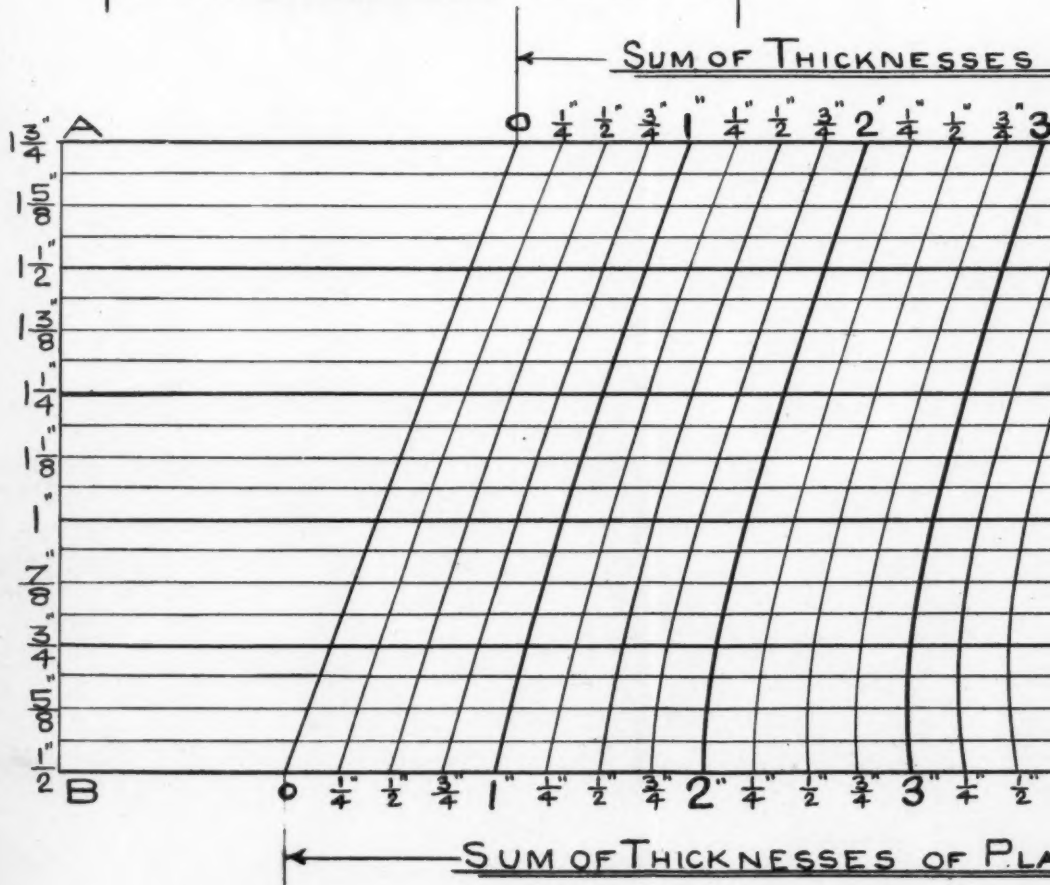
All flanges to be brazed, and those for pipes above 5" diam. to be riveted.

## TO USE DIAGRAM.

*LENGTH TO BE TAKEN WITH A COMMON RULE FROM LINE AT POINT OPPOSITE PROPER DIAM. TO CURVE CORRESPONDING TO SUM OF THICKNESSES OF PLATES.

*THIS LENGTH.

EXAMPLE { 1 1/8" RIVE
MAKE
MEASU



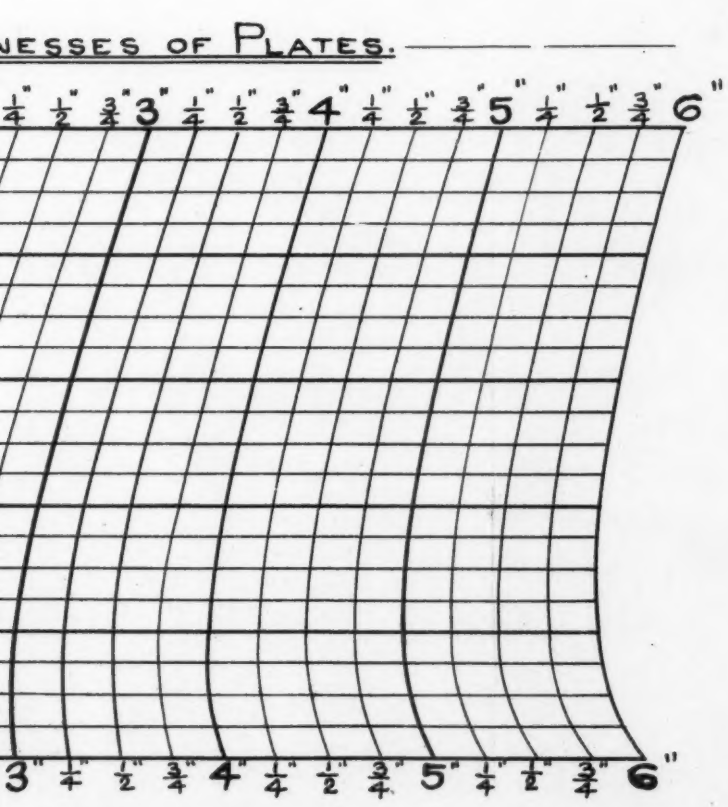

# DIAGRAM FOR LENGTHS

(CONSTRUCTED FOR HOLES DRILLED

DIA. OF RIVETS. MACHINE RIVETTIN

{ $1\frac{1}{8}$" RIVETS TO BE USED, AND PLATES TAKEN TOGETHER MAKE UP $2\frac{3}{4}$": LOOK FOR $1\frac{1}{8}$" ON LINE A B, AND MEASURE FROM IT TO CURVE AT $2\frac{3}{4}$", $5\frac{1}{16}$" REQUIRED.

NESSES OF PLATES.




OF PLATES.

GTHS OF RIVETS.

DRILLED $\frac{1}{16}$" LARGER THAN
IVETTING. BOTH HEADS ALIKE.)

3

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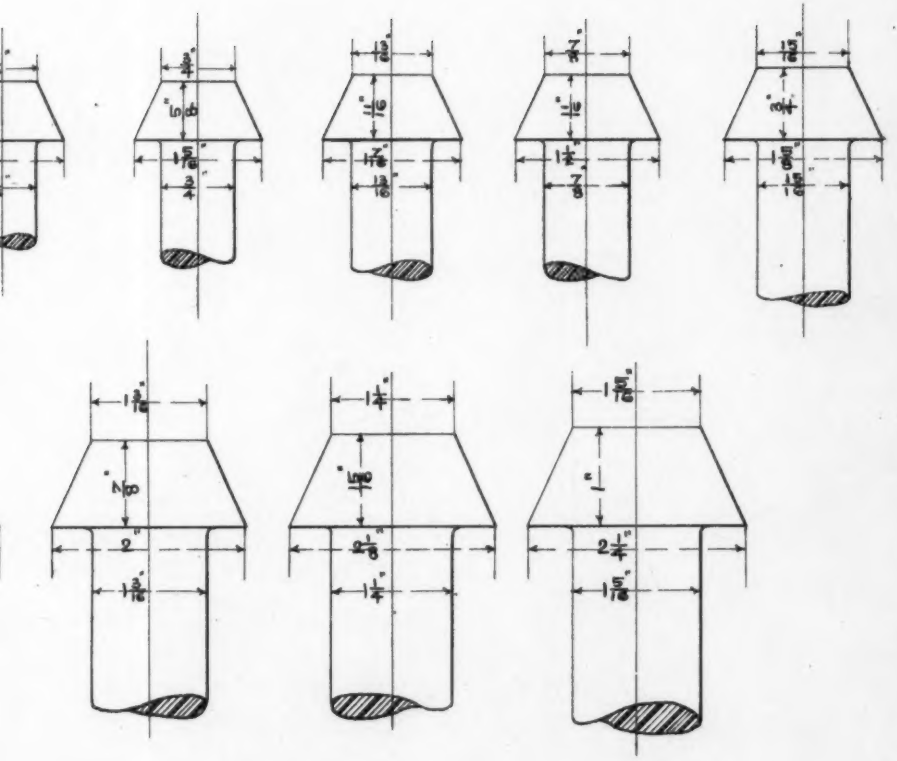

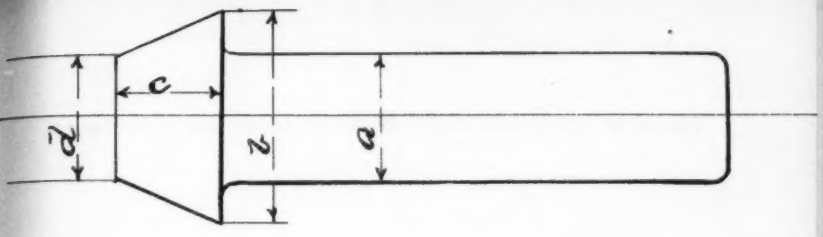


| a | b | c | d | Wt. of 10-Heads. |
|---|---|---|---|---|
| 7/16" | 13/16" | 3/8" | 7/16" | .331 lbs. |
| 1/2" | 15/16" | 7/16" | 1/2" | .531 " |
| 9/16" | 1" | 1/2" | 9/16" | .713 " |
| 5/8" | 1 1/8" | 9/16" | 5/8" | 1.007 " |
| 11/16" | 1 1/4" | 5/8" | 11/16" | 1.373 " |
| 3/4" | 1 5/16" | 5/8" | 3/4" | 1.551 " |
| 13/16" | 1 7/16" | 11/16" | 13/16" | 2.032 " |
| 7/8" | 1 1/2" | 11/16" | 7/8" | 2.258 " |
| 15/16" | 1 5/8" | 3/4" | 15/16" | 2.871 " |
| 1" | 1 3/4" | 13/16" | 1" | 3.584 " |
| 1 1/16" | 1 13/16" | 13/16" | 1 1/16" | 3.91 " |
| 1 1/8" | 1 15/16" | 7/8 | 1 1/8" | 4.761 " |
| 1 3/16" | 2" | 7/8" | 1 3/16" | 5.17 " |
| 1 1/4" | 2 1/8" | 15/16" | 1 1/4" | 6.215 " |
| 1 5/16" | 2 1/4" | 1" | 1 5/16" | 7.391 " |

a
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## XXV.

The preceding dimensions and forms governing copper steam and water pipes and their flanges, and similarly those for boiler rivets, were proposed by the Bureau of Steam Engineering, the object being to secure a uniform standard to be adhered to by the contractors building machinery for the United States Government. The dimensions of rivets proposed and the form of heads and points have met with approval, and that part of the question may be regarded as settled. The thickness of copper pipe proposed, and the thickness and diameter of flanges also may be regarded as a fair solution of a question on which there is much diversity of opinion and practice, as also the number and size of bolts used. A question which is still unsettled is the manner of securing the flange to the pipe. The method proposed was by turning out the end of the pipe into a recess in the flange, and by brazing, or by brazing and riveting combined, as shown by the cuts. The pages of the JOURNAL will be open to makers and users of copper pipes for comments and suggestions, and the final decision will be published whenever it shall have been arrived at.

## XXVI.

The following tables of dimensions and data were compiled at the Bureau of Steam Engineering, and will, no doubt, prove interesting and useful to all the readers of the JOURNAL. They are in a form sufficiently clear to dispense with any explanation.

## XXVII.

## MAXIMUM STEAM JACKET EFFICIENCY.

In the "Journal of the Franklin Institute" for April, 1891, is an article written by Prof. Robert H. Thurston, which is very interesting and goes to show why such varying and apparently irreconcilable results have followed the use of the steam jacket.

(1) IDEAL EFFICIENCY.—The fact is sufficiently well known that the steam jacket as employed on the steam engine, of whatever form and arrangement, is intrinsically a wasteful element, and that its use only gives in certain cases an economical advantage by its repression of wastes of larger magnitude. It checks a serious and unavoidable waste, more or less completely, by a process which as inevitably involves a waste, and which is commonly, but not invariably, a lesser one. The ideal steam engine, such as is treated of in the purely thermo-dynamic study of the steam engine, has a lower efficiency with than it has without a jacket. This is apparent from nine cases which were examined and the efficiencies computed both with and without jacket. The same initial pressure was taken, but varying ratios

In the next issue will be given Table 3.—Speed Trials, which completes the series mentioned under XXVI. Lack of time has prevented its appearance in this number.

Extra copies of these tables may be had by members and subscribers on application to the Secretary-Treasurer. Price, fifty cents.

| | Name of Vessel. | *Philadelphia.* | *San Francisco.* | *Charle* |
|---|---|---|---|---|
| | I.H.P. (designed) | 9,000 | 9,000 | 7,500 |
| | Type | Contract speed, 19 K. Two H. D. A. 3-X. | Contract speed, 19 K. Two H. D. A. 3-X. | Two H. D. |
| | Boiler pressure | 160 | 138 | 95 |
| | Number of revolutions (designed) | ... | ... | 125 |
| | Piston speed | ... | ... | 750 |
| | Diameter of cylinders | 38 58 86 | 42 60 94 | 44 ... |
| | Stroke | 40 | 36 | ... |
| Ratios. | Ratio of H.P. cylinder to I.P. cylinder | 1 to 2.35 | 1 to 2.05 | ... |
| | Ratio of I.P. cylinder to L.P. cylinder | 1 to 2.21 | 1 to 2.47 | ... |
| | Ratio of H.P. cylinder to L.P. cylinder | 1 to 5.21 | 1 to 5.07 | 1 to 3. |
| Cylinders. | Material of cylinders | Cast iron. | Cast iron. | Cast ir |
| | Material of liners | Cast iron. | Cast iron. | Cast ir |
| | Thickness of H.P. cylinder: of liner | 1⅛ 1⅛ | 1⅛ 1⅛ | 1⅛ |
| | Thickness of I.P. cylinder: of liner | 1⅛ 1⅛ | 1⅛ 1⅛ | ... |
| | Thickness of L.P. cylinder: of liner | 1⅛ 1⅛ | 1⅛ 1⅛ | 1⅛ |
| Areas. | Port opening in square inches for steam and exhaust: H.P. | 99 137 177 | 139 143 204 | 150 150 |
| | Port opening in square inches for steam and exhaust: I.P. | 198 275 354 | 270 253 423 | ... ... |
| | Port opening in square inches for steam and exhaust: L.P. | 327 406 571 | 487 489 704 | 307 349 |
| Steam velocities. | Velocity of entering and exhaust steam in feet per second: H.P. | 149 108 83 | 125 118 83 | 117 116 |
| | Velocity of entering and exhaust steam in feet per second: I.P. | 176 127 98 | 131 138 85 | ... ... |
| | Velocity of entering and exhaust steam in feet per second: L.P. | 235 172 135 | 178 176 126 | 215 190 |
| Valves. | Number, type and size of valves, H.P. | One piston, 21 dia. | Two piston, 12 dia. | Two piston |
| | Number, type and size of valves, I.P. | Two piston, 21 dia. | Two piston, 20 dia. | ... |
| | Number, type and size of valves, L.P. | Two piston, 32 dia. | Two slide, 50 × 57. | Two piston |
| | Diameter of valve stems: H.P. : I.P. : L.P. | 3½ 3½ 3½ | 4 4 4 | 3½ |
| | Type of valve gear | Marshall. | Stephenson. | Marsha |
| Steam pipe. | Diameter of main steam pipe | 15¼ | 13.5 | 13 |
| | Thickness of main steam pipe | No. 0 B.W.G. = .34 | No. 3 B.W.G. = .26 | No. 3 B.W. |
| Exhaust pipe. | Diameter of main exhaust pipe | 30 | 30 | 24 |
| | Thickness of main exhaust pipe | No. 7 B.W.G. = .18 | No. 7 B.W.G. = .18 | ... |
| | Mean velocity in main steam pipe | 78 | 118 | 122 |
| | Mean velocity in main exhaust pipe | 109 | 126 | 146 |
| Pistons. | Type of pistons | Conical disc. | Conical disc. | Conical |
| | Material of pistons | Cast steel. | Cast steel.* | Cast st |
| | Thickness (or depth) of piston at center | 2⅜ 2⅜ 2⅜ | 2 2½ 4½ | 2⅜ |
| | Thickness (or depth) of piston at rim | 1½ 1½ 1½ | 1⅜ 1⅜ 7½ | 1⅜ |
| | Breadth of packing ring | 1½ 1½ 1½ | 1¼ 1½ 1½ | 4½ |
| Piston rod. | Material of piston rods | Forged steel. | Forged steel. | Forged s |
| | Diameter of piston rods | 7½ | 7½ | 7 |
| | Maximum thrust on rod per square inch at B.P. | 4,108 | 4,631 | 3,500 |
| Guides. | Material of piston rod guides: of slipper | Cast steel: W. metal. | Forged steel: W. metal. | Cast iron: |
| | Area of slipper for one cylinder | 520 | 607 | 521 |
| | Maximum thrust per square inch of surface | 83.4 | 81.2 | 71. |
| Connecting rod. | Material of connecting rod | Forged steel. | Forged steel. | Forged s |
| | Length of connecting rod, C. to C. | 86 | 72 | 72 |
| | Diameter of connecting rod: necks: center | 6⅜ 7 ... | 6½ 7½ ... | 6½ 7 |
| | Maximum thrust on rod per square inch at B.P. | 5,830 | 6,433 | 4,534 |
| Cross-head. | Number of cross-head journals, one cylinder | Two. | Two. | Two |
| | Dimensions of cross-head journals | 9 dia. × 10 long. | 8½ dia. × 8 long. | 7½ dia. × 8 |
| | Maximum thrust on journals per square inch at B.P. | 1,038 | 1,452 | 1,243 |
| Crank shaft. | Material of crank shaft | Forged steel. | Forged steel. | Forged s |
| | Type of crank shaft | Built up. | Solid 3 sections. | Solid for |
| | Diameter of crank pin: length | 15 16⅝ | 15 16 | 12⅜ |
| | Diameter of crank pin holes: forward: intermediate: aft. | 7½ 7 6½ | 7 7 7 | 4½ ... |
| | Diameter of crank shaft | 14½ | 14½ | 12 |
| | Diameter of crank shaft holes: forward: intermediate: aft. | 7½ 7 6½ | 7 7 7 | 4½ ... |
| | Total length of main journals | 121⅜ | 114 | 87 |
| | Maximum pressure on crank pin per square inch at B.P.*,** | 743 | 823 | 646 |
| | Friction constant for crank pin [f/v]** | 16,195 | 18,162 | 11,937 |
| | Maximum pressure on shaft journals per square inch at B.P*,* | 243 | 365 | 190 |
| | Thickness of crank webs: forward: intermediate: aft. | 9½ 10½ 10½ | 10½ 10½ 10½ | 8 ... |
| Line shaft. | Diameter of line shaft | 13½ | 13½ | 12 |
| | Diameter of hole in line shaft | 6 | 7 | 4 |
| Thrust. | Number of thrust collars | 12 | 11 | 11 |
| | Width of thrust collars | 1⅜ | 1⅜ | 1 |
| | Space between thrust collars | 1⅛ | 2⅛ | 2 |
| | Thrust surface, one shaft | 1,206 | 1,135 | 958 |
| | Thrust surface per I.H.P. (designed) | ... | ... | |

*,* At half stroke. **P = Pressure per square inch on pin at beginning of stroke. V = Circumference of pin in feet × revolutions per minut

# TABLE 1.—ENGINES.

| *rleston.* | *Baltimore.* | *Newark.* | *Yorktown.* | *Concord.* | *Dolphin.* | *Petrel* |
|---|---|---|---|---|---|---|
| 500 | 9,000 | 8,500 | 3,400 | 3,400 | 2,300 | 1,100 |
| D. A. 2-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X. | One V. D. A. 2-X. | One H. B. A |
| 95 | 135 | 160 | 160 | 160 | 90 | 100 |
| 125 | 110 | 115 | 150 | 150 | 75 | 130 |
| 750 | 770 | 766 | 750 | 750 | 600 | 715 |
| ... 85 | 42 60 94 | 34 52 76 | 22 31 50 | 22 31 50 | 42 ... 78 | 25 ... |
| 36 | 42 | 40 | 30 | 30 | 48 | 33 |
| ... | 1 to 2.06 | 1 to 2.35 | 1 to 2 | 1 to 2 | ... | ... |
| ... | 1 to 2.47 | 1 to 2.14 | 1 to 2.61 | 1 to 2.61 | ... | ... |
| o 3.77 | 1 to 5.09 | 1 to 5.01 | 1 to 5.22 | 1 to 5.22 | 1 to 3.44 | 1 to 3.3 |
| t iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iro |
| t iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iro |
| 1⅜ | 1¾ 1½ | 1½ 1½ | 1 1 | 1 1 | 1½ 1½ | 1 1 |
| ... | 1¾ 1½ | 1½ 1½ | 1 1 | 1 1 | ... | ... |
| ... | 2½ 1⅛ | 1½ 1½ | 1 1 | 1 1 | 1½ 1½ | 1 1 |
| 50 219 | 144 144 176 | 107 81 145 | 43 32 54 | 41 33 59 | 105 105 140 | 72 65 |
| ... ... | 252 252 366 | 136 198 257 | 49 68 94 | 57 78 100 | ... ... ... | ... ... |
| 149 559 | 487 487 675 | 272 395 515 | 114 156 211 | 110 129 225 | 216 216 275 | 200 182 |
| 16 80 | 130 130 106 | 120 153 87 | 116 160 91 | 117 146 82 | 132 132 99 | 79 88 |
| ... ... | 153 153 105 | 221 150 116 | 200 146 105 | 168 123 96 | ... ... ... | ... ... |
| 90 120 | 195 195 141 | 237 162 125 | 225 165 122 | 226 193 110 | 221 221 174 | 97 106 |
| ton, 13 dia. | One slide 42 × 42. | One piston, 18 dia. | One piston, 9 dia. | One piston, 9 dia. | One slide, 44 × 49 | One piston, |
| ... | One slide 52 × 64. | One piston, 30 dia. | One piston, 15 dia. | One piston, 15 dia. | ... | ... |
| ton, 26 dia. | One slide 66 × 88. | Two piston, 30 dia. | Two piston, 17 dia. | Two piston, 17 dia. | One slide, 60 × 63 | Two piston, |
| 3½ | 3½ 3½ 4 | 2¾ 3½ 3½ | 2½ 2½ 2½ | 2½ 2½ 2½ | 3½ 3½ | 2¾ |
| shall.†† | Stephenson. | Marshall. | Marshall. | Marshall.†† | Stephenson. | Marshall |
| 13½ | 15 | 12½ | 8 | 8 | 13 | 9 |
| .W.G. = .26 | No. 3 B.W.G. = .26 | No. 2 B.W.G. = .28 | No. 5 B.W.G. = .2 | No. 5 B.W G. = .22 | No. 7 B.W.G. = .18 | No. 5 B.W.G |
| 24 | 30 | 22½ | 18 | 18 | Two 8 × 30. | 16 |
| ... | ... | No. 7 B.W.G. = .18 | No. 10 B.W.G. = .13 | No. 10 B.W.G. = .13 | ... | No. 10 B.W.G |
| 122 | 106 | 103 | 98 | 96 | 105 | 89 |
| 146 | 134 | 161 | 101 | 98 | 99 | 96 |
| cal disc. | Conical disc. | Conical disc. | Conical disc. | Conical disc. | Double shell. | Double sh |
| t steel. | Cast steel. | Cast steel. | Cast steel. | Cast steel. | Composition. | Composit |
| 2¾ | 2 2½ 2½ | 2¼ 2¾ 2¾ | 2¼ 2½ 2½ | 2¼ 2¼ 2¼ | 11 ... 11 | 8 ... |
| 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 9½ ... 9½ | 10½ ... |
| 4½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 7 ... 7 | 2 ... |
| ed steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged st |
| 7½ | 8½ | 6 | 3¾ | 3¾ | 7 | H.P. 4½ L. |
| 500 | 3,296 | 5,139 | 5,529 | 5,529 | 3,240 | 3,067 |
| W. metal. | Cast steel: W. metal. | Cast iron: W. metal. | Cast steel: W. metal. | Cast steel: M. metal.¶ | Cast iron: W. metal. | Cast iron: Com |
| 521 | 621 | 432 | 186 | 186 | 336 | 360 |
| 71.5 | 77.7 | 100 | 78.8 | 78.8 | 84.2 | 35.1 |
| ed steel. | Forged steel. | Wrought iron. | Wrought iron. | Wrought iron. | Forged steel. | Forged st |
| 72 | 84 | 86 | 64 | 64 | 108 | 66 |
| 7 ... | 6 7 ... | 5¾ 6¾ ... | 3¾ 4½ ... | 3¾ 4½ ... | 7 8 7½ | 4½ 5¾ |
| 534 | 6,826 | 5,744 | 5,684 | 5,684 | 3,320 | 3,704 |
| Two. | One. | Two. | Two. | Two. | One. | One. |
| × 8 long. | 8½ dia. × 14 long. | 7½ dia. × 8½ long. | 4½ dia. × 5½ long. | 4½ dia. × 5½ long. | 8 dia. × 11 long. | 5½ dia. × 8 |
| 243 | 1,623 | 1,214 | 1,202 | 1,202 | 1,452 | 1,152 |
| ed steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged st |
| l forged. | Solid forged. | Built up. | Solid forged. | Built up. | Built up. | Solid for |
| 18½ | 15½ 15½ | 13 15 | 9 10 | 9 10 | 13 18 | 9 9 |
| ... 4½ | 7 7 7 | 6 5 4 | 3 3 3 | 3 3 3 | 0 ... 0 | 3 ... |
| 12½ | 15 | 13 | 9 | 9 | 13 | 9 |
| ... 4½ | 7 7 7 | 6 5 4 | 3 3 3 | 3 3 3 | 0 ... 0 | 3 ... |
| 87½ | 114 | 104½ | 71 | 71 | 84 | 35 |
| 646 | 787 | 766 | 695 | 695 | 546 | 470 |
| 937 | 16,633 | 15,498 | 12,909 | 12,740 | 8,524 | 7,802 |
| 190 | 308 | 328 | 240 | 240 | 246 | 144 |
| ... 8 | 11½ 11½ 11½ | 8 9 10 | 6½ 6½ 6½ | 6½ 6½ 7½ | 9 ... 9 | 5½ ... |
| 12½ | 14 | 12½ | 9 | 9 | 13½ | 9 |
| 4½ | 8 | 5 | 3 | 3 | 0 | 3 |
| 11 | 11 | 12 | 9 | 9 | 13 | 8 |
| 1½ | 1½ | 1½ | 1½ | 1½ | 1½ | 1½ |
| 2½ | 2½ | 1½ | 1½ | 1½ | 1½ | 1½ |
| 958 | 1,210 | 1,206.8 | 499 6 | 499.6 | 909.3 | 384 |
| .2554 | .2688 | .2839 | .2940 | .2940 | .3910 | . |

nute. ‡ First figure is for steam at outboard or top end; second for steam at inboard or bottom end; third for exhaust, mean of both ends. * L.P. of comp

# TABLE 1.—ENGINES.

| | *San Francisco.* | *Charleston.* | *Baltimore.* | *Newark.* | *Yorktown.* | *Concord.* |
|---|---|---|---|---|---|---|
| | 9,000 | 7,300 | 9,000 | 8,500 | 3,400 | 3,400 |
| K. | Contract speed, 19 K. | Two H. D. A. 2-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X. | Two H. D. A. 3-X |
| X. | 138 | 95 | 135 | 160 | 160 | 160 |
| | ... | 125 | 110 | 115 | 150 | 150 |
| | ... | 750 | 770 | 766 | 750 | 750 |
| 86 | 42 60 94 | 44 ... 85 | 42 60 94 | 34 52 76 | 22 31 50 | 22 31 |
| | 36 | 36 | 42 | 40 | 30 | 30 |
| | 1 to 2.05 | ... | 1 to 2.06 | 1 to 2.35 | 1 to 2 | 1 to 2 |
| | 1 to 2.47 | ... | 1 to 2.47 | 1 to 2.14 | 1 to 2.61 | 1 to 2.61 |
| | 1 to 5.07 | 1 to 3.77 | 1 to 5.09 | 1 to 5.01 | 1 to 5.22 | 1 to 5.22 |
| | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. |
| | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. |
| | 1½ 1⅜ | 1½ 1⅜ | 1⅝ 1⅜ | 1½ 1½ | 1 1 | 1 1 |
| | 1½ 1½ | ... ... | 1⅝ 1⅝ | 1½ 1½ | 1 1 | 1 1 |
| | 1½ 1½ | 1⅜ ... | 2⅛ 1⅞ | 1½ 1½ | 1 1 | 1 1 |
| 177 | 139 143 204 | 150 150 219 | 144 144 176 | 107 81 145 | 43 31 54 | 41 33 |
| 354 | 270 253 423 | ... ... ... | 252 252 366 | 136 198 257 | 49 68 94 | 57 78 |
| 571 | 487 489 704 | 307 349 559 | 487 487 675 | 272 395 515 | 114 156 211 | 110 129 |
| 83 | 125 118 83 | 117 116 80 | 130 130 106 | 120 153 87 | 116 160 91 | 117 146 |
| 98 | 131 138 85 | ... ... ... | 153 153 105 | 221 150 116 | 200 146 105 | 168 123 |
| 135 | 178 176 126 | 215 190 120 | 195 195 141 | 237 162 125 | 225 165 122 | 226 193 |
| ia. | Two piston, 12 dia. | Two piston, 13 dia. | One slide 42 × 42. | One piston, 18 dia. | One piston, 9 dia. | One piston, 9 dia |
| ia. | Two piston, 20 dia. | ... | One slide 52 × 64. | One piston, 30 dia. | One piston, 15 dia. | One piston, 15 dia |
| ia. | Two slide, 50 × 57. | Two piston, 26 dia. | One slide 66 × 88. | Two piston, 30 dia. | Two piston, 17 dia. | Two piston, 17 dia |
| 3½ | 4 4 4 | 3½ Marshall.†† 3½ | 3½ Stephenson. 4 | 2⅝ 3⅜ 3½ | 2½ 2½ 2½ | 2½ 2½ |
| | Stephenson. | | | Marshall. | Marshall. | Marshall.†† |
| | 13.5 | 13⅛ | 15 | 12½ | 8 | 8 |
| .34 | No. 3 B W.G. = .26 | No. 3 B.W.G. = .26 | No. 3 B.W.G. = .26 | No. 2 B.W.G. = .28 | No. 5 B.W.G. = .2 | No. 5 B W G. = |
| | 30 | 24 | 30 | 22½ | 18 | 18 |
| .18 | No. 7 B.W.G. = .18 | ... | ... | No. 7 B.W.G. = .18 | No. 10 B.W.G. = .13 | No. 10 B.W.G. = |
| | 118 | 122 | 106 | 103 | 98 | 96 |
| | 126 | 146 | 134 | 161 | 101 | 98 |
| | Conical disc. | Conical disc. | Conical disc. | Conical disc. | Conical disc. | Conical disc. |
| | Cast steel.* | Cast steel. | Cast steel. | Cast steel. | Cast steel. | Cast steel. |
| 2⅜ | 2 2½ 4½ | 2⅜ 2⅜ | 2 2½ 2½ | 2⅜ 2⅜ 2⅜ | 2½ 2½ 2½ | 2½ 2½ |
| 1½ | 1⅜ 1⅜ 7⅜ | 1⅜ 1½ | 1½ 1⅜ 1⅜ | 1½ 1½ 1½ | 1⅜ 1⅜ 1⅜ | 1⅜ 1⅜ |
| 1½ | 1½ 1½ 1½ | 4½ 4½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ 1½ | 1½ 1½ |
| | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. |
| | 7½ | 7½ | 8½ | 6 | 3⅜ | 3⅜ |
| | 4,621 | 3,500 | 3,296 | 5,139 | 5,529 | 5,529 |
| etal. | Forged steel : W. metal. | Cast iron : W. metal. | Cast steel : W. metal. | Cast iron : W. metal. | Cast steel : W. metal. | Cast steel : M. meta |
| | 607 | 521 | 621 | 432 | 186 | 186 |
| | 81.2 | 71.5 | 77.7 | 100 | 78.8 | 78.8 |
| | Forged steel. | Forged steel. | Forged steel. | Wrought iron. | Wrought iron. | Wrought iron. |
| | 72 | 72 | 84 | 86 | 64 | 64 |
| ... | 6½ 7½ ... | 6½ 7 ... | 6 7 ... | 5⅞ 6⅞ ... | 3⅜ 4½ ... | 3⅜ 4½ |
| | 6,433 | 4,534 | 6,826 | 5,744 | 5,684 | 5,684 |
| g. | Two. | Two. | One. | Two. | Two. | Two. |
| | 8½ dia. × 8 long. | 7½ dia. × 8 long. | 8½ dia. × 14 long. | 7½ dia. × 8½ long. | 4½ dia. × 5½ long. | 4½ dia. × 5½ long |
| | 1,452 | 1,243 | 1,623 | 1,214 | 1,202 | 1,202 |
| | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. |
| | Solid 3 sections. | Solid forged. | Solid forged. | Built up. | Solid forged. | Built up. |
| | 15 16 | 12⅜ 18½ | 15½ 15⅝ | 13 15 | 9 10 | 9 10 |
| 6½ | 7 7 7 | 4½ ... 4½ | 7 7 7 | 6 5 4 | 3 3 3 | 3 3 |
| | 14½ | 12½ | 15 | 13 | 9 | 9 |
| 6½ | 7 7 7 | 4½ ... 4½ | 7 7 7 | 6 5 4 | 3 3 3 | 3 3 |
| | 114 | 87½ | 114 | 104½ | 71 | 71 |
| | 823 | 646 | 787 | 766 | 695 | 695 |
| | 18,162 | 11,937 | 16,623 | 15,498 | 12,909 | 12,740 |
| | 365 | 190 | 308 | 328 | 240 | 240 |
| 10½ | 10½ 10½ 10½ | 8 ... 8 | 11¼ 11½ 11¼ | 8 9 10 | 6½ 6½ 6½ | 6½ 6½ |
| | 12½ | 12½ | 14 | 12½ | 9 | 9 |
| | 7 | 4½ | 8 | 5 | 3 | 3 |
| | 11 | 11 | 11 | 12 | 9 | 9 |
| | 1⅜ | 1½ | 1⅝ | 1⅜ | 1½ | 1½ |
| | 2½ | 2½ | 2⅝ | 1½ | 1⅜ | 1⅜ |
| | 1,135 | 958 | 1,210 | 1,206.8 | 499 6 | 499.6 |
| | ... | .2554 | .2688 | .2839 | .2940 | .2940 |

mference of pin in feet × revolutions per minute. ‡ First figure is for steam at outboard or top end; second for steam at inboard or bottom end; third for

| | *Dolphin.* | *Petrel.* | *Vesuvius.* | *Chicago.* | *Boston.* |
|---|---|---|---|---|---|
| | 2,300 | 1,100 | 3,200 | 5,000 | 3,500 |
| 3-X. | One V. D. A. 2-X. | One H. B. A. 2-X. | Two V. D. A. 3-X. | Two beam 2-X. | One H. B. A. 2-X. |
| | 90 | 100 | 160 | 100 | 90 |
| | 75 | 130 | ... | 64 | 70 |
| | 600 | 715 | ... | 608 | 490 |
| 50 | 42 ... 78 | 25 ... 46 | 21½ 31 34 34 | 45 ... 78 | 54 74 74 |
| | 48 | 33 | 20 | 57 | 42 |
| | ... | ... | 1 to 2.07 | ... | ... |
| | ... | ... | 1 to 2.40 | ... | ... |
| | 1 to 3.44 | 1 to 3.38 | 1 to 5.00 | 1 to 3 | 1 to 3.75 |
| | Cast iron. | Cast iron. | Cast iron. | Cast iron. | Cast iron. |
| | Cast iron. | Cast iron. | ... | Cast iron. | Cast iron. |
| | 1½ 1½ | 1 1 | ⅞ ... | 1½ 1½ | 1⅜ 1½ |
| | ... | ... ... | ¾ ... | ... ... | ... ... |
| | 1½ 1½ | 1 1 | ⅝ ... | 1½ 1½ | 1½ 1½ |
| 59 | 105 105 140 | 72 65 72 | 46 35 59 | 119 133 140 | 134 134 189 |
| 100 | ... ... ... | ... ... ... | 63 86 116 | ... ... ... | ... ... ... |
| 225 | 216 216 275 | 200 182 200 | 63 86 116 | 237 237 300 | 191 187 285 |
| 82 | 132 132 99 | 79 88 79 | 117 155 92 | 105 131 124 | 144 144 102 |
| 96 | ... ... ... | ... ... ... | 179 131 97 | ... ... ... | ... ... ... |
| 110 | 221 221 174 | 97 106 97 | 214 158 117 | 221 221 175 | 190 194 127 |
| dia. | One slide, 44 × 49 | One piston, 15 dia. | One piston, 11 dia. | One slide, 43 × 51 | Two slide, 28 × 46 |
| ; dia. | ... | ... ... ... | One piston, 18 dia. | ... | ... |
| r dia. | One slide, 60 × 63 | Two piston, 16 dia. | One piston, 18 dia. | One slide, 59 × 63 | Two slide, 37 × 46 |
| 2½ | 3½ 3½ | 2¾ 2¾ | 2½ 2½ 2½ | 3 (2) 3 (2) | 2¾ 2¾ |
| † | Stephenson. | Marshall.†† | Marshall. | Stephenson.†† | Stephenson.†† |
| | 13 | 9 | 9 | 13 | 17 |
| = .22 | No. 7 B.W.G. = .18 | No. 5 B.W.G. = .22 | No. 5 B.W.G. = .22 | No. 2 B.W.G. = .28 | No. 3 B.W.G. = .26 |
| = .13 | Two 8 × 30. | 16 | 13½ | 22 | 19 |
| | ... | No. 10 B.W.G. = .13 | No. 10 B.W.G. = .13 | No. 5 B.W.G. = .22 | No. 10 B.W.G. = .13 |
| | 105 | 89 | 85 | 131 | 84 |
| c. | 99 | 96 | 95 | 155 | 128 |
| | Double shell. | Double shell. | Conical disc. | Double shell. | Double shell. |
| | Composition. | Composition. | Cast steel. | Composition. | Composition. |
| 2½ | 11 ... 11 | 8 ... 10½ | 1½ 1½ 1½ | 11⅜ ... 11⅜ | 13 ... 15 |
| 1¾ | 9½ ... 9½ | 10½ ... 10½ | ⅜ 1 1 | 7½ ... 7½ | 15 ... 15 |
| 2½ | 7 ... 7 | 2 ... 2 | 1½ 1½ 1½ | 5½ ... 5½ | 12 ... 12 |
| l. | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Forged steel. |
| | 7 | H.P. 4½ L.P. 3½ (2) | 3½ | 7½ | H.P. 8 L.P. 6½ (2) |
| | 3,240 | 3,067 | 6,037 | 3,852 | 4,100 |
| metal.¶ | Cast iron: W. metal. | Cast iron: Composition. | Cast steel: W. metal. | Cast iron: Composition. | Cast iron: Composition. |
| | 336 | 360 | 105 | 238 | 483 |
| n. | 84.2 | 35.1 | 126.1 ‖ | 182.6 | 102.4 |
| | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Wrought iron. |
| | 108 | 66 | 45 | 108 | 90 |
| ... | 7 8 7½ | 4½ 5½ 4½ | 3 4 ... | 8 9 9½ | 8 8½ 9 |
| | 3,320 | 3,704 | 8,432 | 3,307 | 4,237 |
| ong. | One. | One. | One. | Two. | One. |
| l. | 8 dia. × 11 long. | 5½ dia. × 8 long. | 4½ dia. × 6 long. | 7½ dia. × 8 long. | 10 dia. × 14 long. |
| | 1,452 | 1,152 | 2,205 | 1,378 | 1,513 |
| | Forged steel. | Forged steel. | Forged steel. | Forged steel. | Wrought iron. |
| | Built up. | Solid forged. | Solid forged. | Built up. | Built up. |
| | 13 18 | 9 12 | 7½ 9 | 14½ 19 | 15 18 |
| 3 | 0 ... 0 | 3 ... 3 | 4 2 2 | 0 ... 0 | 0 ... 0 |
| | 13 | 9 | 7½ | 14½ | 16 |
| 3 | 0 ... 0 | 3 ... 3 | 4 2 2 | 0 ... 0 | 0 ... 0 |
| | 84 | 55 | 68 | 94 | 156 |
| | 546 | 470 | 850 | 601 | 784 |
| | 8,524 | 7,802 | 19,424 | 9,335 | 12,823 |
| | 246 | 144 | 293 | 260 | 254 |
| 7½ | 9 ... 9 | 5½ ... 5½ | 5½ 5½ 5½ | 10½ ... 10½ | 9 ... 9 |
| | 13½ | 9 | 7½ | 14½ | 16 |
| | 0 | 3 | 3 | 0 | 0 |
| | 13 | 8 | 11 | 15 | 19 |
| | 1⅜ | 1½ | 1½ | 1⅜ | 2½ |
| | 1½ | 1⅜ | 1½ | 1½ | 1½ |
| | 909.3 | 384 | 518 | 988 | 2,615 |
| 0 | .3910 | .3490 | .3237 | .3952 | .7471 |

for exhaust, mean of both ends. * L.P. of composition. ‖ Double shell. † Full stroke. ¶ Magnolia metal. †† Modified gear.

| | Name of Vessel. | *Philadelphia.* | *San Francisco.* | *Charleston* |
|---|---|---|---|---|
| Condenser. | Total length of condenser | 14–1½ | 12–9½ | 15–4 |
| | Diameter of shell inside | 5–8 | 6–6 | 5–7½ |
| | Material and thickness of shell | Composition: ⅜ | Composition: ⅜ | Copper: |
| | Length of tubes exposed | 12 | 9–11⅞ | 13–4½ |
| | Diameter of tubes outside | ⅝ | ⅝ | ⅝ |
| | Number of tubes in one condenser | 3,440 | 4,453 | 3,092 |
| | Cooling surface in one condenser | 6,755 | 7,259 | 6,751 |
| Air pump. | Type of air pump | V. S. A. independent. | H. S. A. independent. | V. S. A. indepe |
| | Diameter of air pump steam cylinder: H.P.: L.P. | 11 18 | 12 19 | Two: |
| | Diameter of air pump water cylinder | Two: 24 | Two: 26 | Two: |
| | Stroke of air pump | 18 | 18 | 16 |
| | Material of pump rod | Composition. | Aluminum bronze. | Aluminum bro |
| | Diameter of pump rod | 2½ | 2½ | 3½ |
| Circulating pump | Type of circulating pump | Centrifugal. | Centrifugal. | Centrifugal |
| | Diameter of circulating pump steam cylinder: H.P.: L.P. | ... 11 | 7½ 12 | ... |
| | Diameter of circulating pump water cylinder | ... | ... | ... |
| | Stroke of circulating pump | 6 | 6 | 8 |
| | Diameter of wheel or runner | 2–8 | 3–0 | 2–8 |
| | Diameter of suction | 15 | 17 | 14 |
| Propeller. | Type of propeller | Modified Griffiths. | Modified Griffiths. | Modified Grif |
| | Material of propeller | Manganese bronze. | Manganese bronze. | Manganese bro |
| | Diameter of propeller | 14–6 | 13–6 | 14–0 |
| | Mean pitch of propeller | 20–4½ | 18–9 | 17–6 |
| | Number of blades | Three. | Three. | Three. |
| | Helicoidal area | 57.17 | 57.59 | 54.75 |
| | Projected area | 43.86 | 44.55 | 43.35 |
| | Greatest width of blade: at radius | 53½ 3–11 | 55½ 3–3 | 49½ |
| | Thickness of blade at hub | 6½ | 5½ | 5½ |
| | Diameter of hub | 3–11½ | 3–11 | 3–6 |
| | Length of hub | 3–1½ | 3–1 | 2–5 |
| | Ratio of diameter to pitch | 1 to 1.406 | 1 to 1.39 | 1 to 1.25 |
| | Ratio of blade disc area to immersed mid section | 1 to 2.470 | 1 to 2.692 | 1 to 2.405 |
| Boilers. | Number and type of main boilers | Four D.E. Scotch. | Four D.E. Scotch. | Six S.E. Lo |
| | Length of double-ended boilers: single-ended boilers | 19–10⅞ ... | 19–2 ... | ... |
| | Diameter of double-ended boilers: single-ended boilers | 14–0 ... | 14–8 ... | For'd, 11–0: Aft |
| | Thickness of shell: double-ended boilers: single-ended boilers. | 1⅛ ... | 1 15/32 ... | ... |
| | Material of boilers | Steel. | Steel. | Steel. |
| Furnaces. | Total number and type of furnaces | 32 Corrugated. | 24 Corrugated. | 18 Addison |
| | Number of furnaces in each D.E. boiler; S.E. boiler | Eight: ... | Six: ... | ... T |
| | Least internal diameter of furnace | 36 | 42 | For'd, 42½: Af |
| | Length of grate | 6–6 | 6–7 | 6–1½ |
| | Total grate surface [main boilers] | 624 | 553 | 421.2 |
| Tubes. | Material of tubes | Wrought iron. | Wrought iron. | Wrought iro |
| | Outside diameter of tubes | 2½ | 2½ | 3 |
| | Length between tube sheets | 7–7½ | 6–9 | 7–1⅛ |
| | Total number of ordinary tubes | 3,184 | 3,308 | 1,956 |
| | Total number of stay tubes | 672 | 896 | 342 |
| | Thickness of ordinary tubes | No. 12 B.W.G. | No. 12 B.W.G. | No. 10 B.W. |
| | Thickness of stay tubes | No. 8 B.W.G. | No. 6 B.W.G. | ⅜ |
| | Pitch of tubes: horizontal: vertical | 3⅛ 3½ | 3½ 3½ | Staggered: |
| | Calorimeter of tubes | 85.3 | 91.56 | 90.12 |
| | Depth of combustion chamber | 4–6 | 2–5 | 4–6 |
| Thickness. | Thickness of tube sheets | ⅝ | ⅝ | 1⅛ |
| | Thickness of front head above tube sheet | 15/16 | 13/16 | ⅞ |
| | Thickness of combustion chamber | ⅞ | ⅞ | ⅞ |
| | Thickness of furnace | 11/16 | ½ | 1⅛ |
| | Material and number of braces above tubes | ½ | ½ | ½ |
| | Diameter of braces above tubes | Steel: 18 | Steel: 19 | Steel: |
| | Total heating surface | 2½ | 2½ and 2½ | 2½ |
| | Area of water surface: inches above tubes | 20,457.68 | 19,657.56 | 15,147 |
| | Total steam room [main boilers] | 948 6¾ | 960 8 | 1,185 |
| Ratios. | Steam space divided by grate area | 2,348 | 2,689.6 | 3,075 |
| | Heating surface divided by grate surface | 3.76 | 4.86 | 7.3 |
| | Grate surface divided by calorimeter of tubes | 32.78 | 35.55 | 35.96 |
| | Grate surface divided by calorimeter of smoke-pipe | 7.3 | 6.06 | 4.67 |
| | Capacity of furnaces and combustion chambers divided by G.S. | 8.72 | 8.19 | 4.63 |
| | Weight of one boiler with water: D.E.: S.E. | 3.35 | 3.40 | 4.15 |
| | Total weight in boiler comp't per 100 square feet of H.S. | 87.9 ... | 91.23 ... | For'd, 52.84: Af |
| | | 2.39 tons. | 2.279 tons. | 2.198 tons. |

BLE 2.—AUXILIARIES AND BOILERS.

| ston. | *Baltimore.* | *Newark.* | *Yorktown.* | *Concord.* | *Dolphin.* | *Petrel.* |
|---|---|---|---|---|---|---|
| | 11-7 | 13-11½ | 8-11 | 8-11 | 11-6 | 12-5 |
| | 5-4 × 7-4 | 5-8 | 4-5½ | 4-5½ | 6-10 × 3-7 | 3-6 |
| ½ ½ | Composition: ½ | Composition: ½ | Rolled brass: ½ | Rolled brass: ½ | Cast iron: 1½ | Rolled brass: |
| | 7-6 | 12-0 | 7-0 | 7-0 | 9-0 | 11-0 |
| | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| | 4,256 | 3,186 | 2,080 | 2,151 | 2,264 | 1,370 |
| | 6,184.3 | 6,254 | 2,380 | 2,463 | 4,000 | 2,466 |
| lependent. | H. D. A. dependent. | V. S. A. independent. | V. S. A. independent. | V. S. A. Davidson. | V. S. A. Blake. | Incl'd S.A. depen |
| 10 | ... ... | 11 18 | 11 18 | One: 14 | One: 22 | ... |
| 24 | One: 28 | Two: 24 | One: 24 | Two: 18 | Two: 24½ | One: |
| | 42 | 18 | 12 | 12 | 21 | 18 |
| bronze. | Muntz metal. | Composition. | Steel, compos'n cased. | Wrought iron, in trunk. | Phosphor bronze. | Rolled phos. bron |
| | 4 | 2½ | 2½ | 1½ | 3 | 2½ |
| ugal. | Centrifugal. | Centrifugal. | Vide air pump. | H. D. A. Vide air pump. | H. D. A. Vide air pump. | Centrifugal. |
| 9 | ... 11 | ... 10 | ... ... | ... 14 | ... 22 | ... |
| | ... | ... | 20 | 18 | 26 | ... |
| | 12 | 6 | 12 | 12 | 24 | 6 |
| | 3-6 | 2-8 | ... | ... | ... | 2-0 |
| | 17 | 15 | 11 | 11 | 14 | 10 |
| Griffiths. | Modified Griffiths. | Modified Griffiths. | Modified Griffiths. | Modified Griffiths. | Hirsch. | Modified Griffith |
| bronze. | Manganese bronze. | Manganese bronze. | Manganese bronze. | Manganese bronze. | Cast iron. | Manganese bron |
| | 14-6 | 14-6 | 10-6 | 10-6 | 14-3 | 9-9 |
| | 20-0 | 18-11⅜ | 12-6 | 13-2½ | 24-1 | 12-3 |
| e. | Three. | Three. | Three. | Three. | Four. | Three. |
| 35 | 57.17 | 52.77 | 25.42 | 26.52 | 72.14 | 23.55 |
| 4-0 | 45. | 40. | 20.6 | 21.45 | 48.56 | 17.75 |
| ½ | 53½ 3-11 | 48 3-11 | 29½ 3-3 | 29½ 3-3 | 40½ 3-7 | 28 2- |
| | 6½ | 6½ | 4 | 4 | 7 | 3 |
| | 3-11⅛ | 3-11⅛ | 2-6½ | 2-6½ | 2-5 | 10 |
| | 3-1⅜ | 3-1⅜ | 22½ | 22½ | 2-5 | 22 |
| .25 | 1 to 1.38 | 1 to 1.308 | 1 to 1.19 | 1 to 1 257 | 1 to 1.689 | 1 to 1.28 |
| .405 | 1 to 2.474 | 1 to 2.350 | 1 to 2.516 | 1 to 2.542 | 1 to 2.232 | 1 to 3.83 |
| Low. | Four D.E. Scotch. | Four D.E. Scotch. | Four S.E. Low. | Four S.E. Low. | 2 D.E., 2 S.E., Scotch. | Two S.E. Low |
| 19-3 | 17-8 ... | 19-4½ ... | ... 17-9 | ... 17-9 | 18-3 9-6 | ... 18- |
| Aft, 11-6 | 14-8 ... | 13-6 ... | ... 9-9 | ... 9-9 | 11-0 11-0 | ... 8- |
| 1¼ | 1⅛ ... | 1⅛ ... | ... 1⅛ | ... 1⅛ | ¾ ¾ | ... |
| | Steel. | Steel. | Steel. | Steel. | Steel. | Steel. |
| son ring. | 32 Corrugated. | 24 Corrugated. | 12 Corrugated. | 12 Corrugated. | 19 Addison ring. | 4 Corrugat |
| Three. | Eight: ... | Six: ... | ... Three. | ... Three. | Four: Two. | ... T |
| Aft, 44½ | 36 | 43 | 37 | 37 | 40 | 40 |
| ½ | 6-10 | 6-0 | 5-9 | 5-9 | 6-9 | 7-0 |
| 2 | 656 | 540 | 220 | 220 | 270 | 93.2 |
| t iron. | Wrought iron. | Wrought iron. | Wrought iron. | Wrought iron. | Drawn brass. | Wrought iron |
| | 2½ | 2½ | 2½ | 2½ | 2¾ | 2⅞ |
| | 6-4 | 7-6 | 6-11 | 6-11 | 6-1⅛ | 7-10 |
| | 2,848 | 2,416 | 1,480 | 1,480 | 1,200 | 352 |
| | 944 | 528 | 272 | 272 | 108 | 44 |
| .W.G. | No. 8 B.W.G. | No. 12 B.W.G. | No. 11 B.W.G. | No. 11 B.W.G. | No. 11 B.W.G. | No. 11 B.W.G |
| ⅛ | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ |
| 4½ | 3½ 3½ | 3½ 3½ | 3⅜ 3½ | 3⅜ 3½ | 3⅜ 3½ | 4½ |
| 12 | 86.72 | 81.8 | 37.68 | 37.68 | 44.80 | 15.4 |
| | 2-1½ | 4-6 | 3-9 | 3-9 | 2-6 | 9 inche |
| | ⅜ ⅛ | ⅜ ⅛ | ⅜ ⅛ | ⅜ ⅛ | ⅜ F.C. ⅜ B.C. | ⅜ |
| | ¼ | ¼ | ¼ | ¼ | ⅜ D.E. ⅜ S.E. | ¼ |
| | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ |
| | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ |
| | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ | ⅛ |
| 18 | Steel: 32 | Steel: 20 | Steel: 15 | Steel: 15 | Steel: 21 | Steel: |
| | 2 | 2½ | 2½ | 2½ | 1⅜ | 2½ |
| | 16,674.7 | 16,739.8 | 8,091.88 | 8,209.6 | 6,550.8 | 2,796 |
| 8½ | 932 6 | 1,009.6 6½ | 606 10 | 612 8 | 523.2 10 | 176 |
| | 3,012 | 2,053.8 | 1,194 | 1,311 | 1,261.8 | 594 |
| 3 | 4.59 | 3.8 | 6 | 5.96 | 4.67 | 6.37 |
| 96 | 25.42 | 30.99 | 36.78 | 37.3 | 24.26 | 30 |
| 67 | 7.56 | 6.6 | 5.8 | 5 8 | 6.02 | 6.05 |
| 63 | 7 45 | 8.14 | 5.7 | 5.7 | 6.54 | 5.86 |
| 15 | 2 97 | 3.72 | 2.95 | 2.95 | 3.85 | 2.48 |
| Aft, 58.15 | 81 ... | 80.12 ... | ... 41.1 | ... 41.66 | 54.0 27.3 | ... |
| ons. | 2.94 tons. | 2.37 tons. | 2.35 tons. | 2.68 tons. | 3.10 tons. | 2.365 tons. |

* Approximate.

# TABLE 2.—AUXILIARIES AND BOI

| *Philadelphia.* | *San Francisco.* | *Charleston.* | *Baltimore.* | *Newark.* | *Yorktown.* |
|---|---|---|---|---|---|
| 14–1½ | 12–9½ | 15–4 | 11–7 | 13–11½ | 8–11 |
| 5–8 | 6–6 | 5–7½ | 5–4 × 7–4 | 5–8 | 4–5½ |
| Composition : ½ | Composition : ½ | Copper : ½ | Composition : ½ | Composition : ½ | Rolled brass : ½ |
| 12 | 9–11⅞ | 13–4½ | 7–6 | 12–0 | 7–0 |
| ⅝ | ⅝ | ⅝ | ⅝ | ⅝ | ⅝ |
| 3,440 | 4,453 | 3,092 | 4,256 | 3,186 | 2080 |
| 6,755 | 7,259 | 6,751 | 6,184.3 | 6,254 | 4380 |
| V. S. A. independent. | H. S. A. independent. | V. S. A. independent. | H. D. A. dependent. | V. S. A. independent. | V. S. A. independent. |
| 11 18 | 12 19 | Two : 10 | ... ... | 11 18 | 11 18 |
| Two : 24 | Two : 26 | Two : 24 | One : 28 | Two : 24 | One : 24 |
| 18 | 18 | 16 | 42 | 18 | 12 |
| Composition. | Aluminum bronze. | Aluminum bronze. | Muntz metal. | Composition. | Steel, compos'n cased. |
| 2½ | 2½ | 3½ | 4 | 2½ | 2½ |
| Centrifugal. | Centrifugal. | Centrifugal. | Centrifugal. | Centrifugal. | Wide air pump. |
| ... 11 | 7½ 12 | ... 9 | ... 11 | ... 10 | ... ... |
| ... | ... | ... | ... | ... | 20 |
| 6 | 6 | 8 | 12 | 6 | 12 |
| 2–8 | 3–0 | 2–8 | 3–6 | 2–8 | ... |
| 15 | 17 | 14 | 17 | 15 | 11 |
| Modified Griffiths. | Modified Griffiths. | Modified Griffiths. | Modified Griffiths. | Modified Griffiths. | Modified Griffiths. |
| Manganese bronze. | Manganese bronze. | Manganese bronze. | Manganese bronze. | Manganese bronze. | Manganese bronze. |
| 14–6 | 13–6 | 14–0 | 14–6 | 14–6 | 10–6 |
| 20–4½ | 18–9 | 17–6 | 20–0 | 18–11½ | 12–6 |
| Three. | Three. | Three. | Three. | Three. | Three. |
| 57.17 | 57.59 | 54.75 | 57.17 | 52.77 | 25.42 |
| 43.86 | 44.55 | 43.35 | 45. | 40. | 20.6 |
| 53½ 3–11 | 55½ 3–3 | 49½ 4–0 | 53½ 3–11 | 48 3–11 | 29½ 3–3 |
| 6½ | 5½ | 5½ | 6½ | 6½ | 4 |
| 3–11½ | 3–11 | 3–6 | 3–11½ | 3–11½ | 2–6½ |
| 3–1½ | 3–1 | 2–5 | 3–1½ | 3–1½ | 22½ |
| 1 to 1.406 | 1 to 1.39 | 1 to 1.25 | 1 to 1.38 | 1 to 1.308 | 1 to 1.19 |
| 1 to 2.470 | 1 to 2.692 | 1 to 2.405 | 1 to 2.474 | 1 to 2.350 | 1 to 2.516 |
| Four D.E. Scotch. | Four D.E. Scotch. | Six S.E. Low. | Four D.E. Scotch. | Four D.E. Scotch. | Four S.E. Low. |
| 19–10⅝ ... | 19–2 ... | ... 19–3 | 17–8 ... | 19–4⅞ ... | ... 17–9 |
| 14–0 ... | 14–8 ... | For'd, 11–0 : Aft, 11–6 | 14–8 ... | 13–6 ... | ... 9–9 |
| 1½ ... | 1⅞ ... | ... 1¼ | 1½ ... | 1½ ... | ... 1⅛ |
| Steel. | Steel. | Steel. | Steel. | Steel. | Steel. |
| 32 Corrugated. | 24 Corrugated. | 18 Addison ring. | 32 Corrugated. | 24 Corrugated. | 12 Corrugated. |
| Eight : ... | Six : ... | ... Three. | Eight : ... | Six : ... | ... Three. |
| 36 | 42 | For'd, 42½ : Aft, 44½ | 36 | 43 | 37 |
| 6–6 | 6–7 | 6–1½ | 6–10 | 6–0 | 5–9 |
| 624 | 553 | 421.2 | 636 | 540 | 220 |
| Wrought iron. | Wrought iron. | Wrought iron. | Wrought iron. | Wrought iron. | Wrought iron. |
| 2½ | 2½ | 3 | 2½ | 2½ | 2½ |
| 7–7½ | 6–9 | 7–1⅝ | 6–4 | 7–6 | 6–11 |
| 3,184 | 3,308 | 1,956 | 2,848 | 2,416 | 1,480 |
| 672 | 896 | 342 | 944 | 528 | 272 |
| No. 12 B.W.G. | No. 12 B.W.G. | No. 10 B.W.G. | No. 8 B.W.G. | No. 12 B.W.G. | No. 11 B.W.G. |
| No. 8 B.W.G. | No. 6 B.W.G. | ⅞ | ½ | No. 7 B.W.G. | No. 6 B.W.G. |
| 3½ 3½ | 3½ 3½ | Staggered : 4½ | 3½ 3½ | 3½ 3½ | 3½ 3½ |
| 85.3 | 91.56 | 90.12 | 86.72 | 81.8 | 37.68 |
| 4–6 | 2–5 | 4–6 | 2–1½ | 4–6 | 3–9 |
| ⅞ | ¾ | 1⅛ | ⅝ | ⅞ | ⅞ |
| ⅝ | ½ | ¾ | ½ | ⅝ | ⅝ |
| 1⅛ | 1⅛ | 1⅛ | ⅜ | 1⅛ | ½ |
| ½ | ½ | ½ | ⅜ | ½ | ⅝ |
| Steel : 18 | Steel : 19 | Steel : 18 | Steel : 32 | Steel : 20 | Steel : 15 |
| 2½ | 2½ and 2½ | 2½ | 2 | 2½ | 2½ |
| 20,457.68 | 19,657.56 | 25,147 | 16,674.7 | 16,739.8 | 8,091.88 |
| 948 6½ | 960 8 | 1,185 8½ | 932 6 | 1,009.6 6½ | 606 10 6 |
| 2,348 | 2,689.6 | 3,075 | 3,012 | 2,053.8 | 1,194 |
| 3.76 | 4.86 | 7.3 | 4.59 | 3.8 | 6 |
| 32.78 | 35.55 | 35.96 | 25.42 | 30.99 | 36.78 |
| 7.3 | 6.06 | 4.67 | 7.56 | 6.6 | 5.8 |
| 8.72 | 8.19 | 4.63 | 7 45 | 8.14 | 5.7 |
| 3.35 | 3 40 | 4.15 | 2 97 | 3.72 | 2.95 |
| 87.9 ... | 91.23 ... | For'd, 52.84 : Aft, 58.15 | 81 ... | 80.12 ... | ... 41.1 |
| 2.39 tons. | 2.279 tons. | 2.198 tons. | 2.94 tons. | 2.37 tons. | 2.35 tons. |

* Approximate.

| *Concord.* | *Dolphin.* | *Petrel.* | *Vesuvius.* | *Chicago.* | *Boston.* |
|---|---|---|---|---|---|
| 8–11 | 11–6 | 12–5 | 8–3 | 14–7 | Two 12–4½ |
| 4–5½ | 6–10 × 3–7 | 3–6 | 4–4½ | 9–11 × 2–8½ | 3–9½ × 8–6 |
| Rolled brass: ¼ | Cast iron: 1⅛ | Rolled brass: ¼ | Composition: ⅜ | Cast iron: 1¼ | Cast iron: 1 |
| 7–0 | 9–0 | 11–0 | 6 6 | 12–0 | 6–6 |
| ⅜ | ⅜ | ⅜ | ⅜ | ⅜ | ⅜ |
| 2,151 | 2,264 | 1,370 | 2,124 | 1,668 | 4,140 |
| 2,463 | 4,000 | 2,466 | 2,259 | 4,587 | 4,403 |
| V. S. A. Davidson. | V. S. A. Blake. | Incl'd S.A. dependent. | V. S. A. dependent. | V. S. A. Blake. | H. D. A. independent. |
| One: 14 | One: 22 | ... ... | ... ... | One: 22 | One: 18 |
| Two: 18 | Two: 24½ | One: 14 | Two: 6⅞ 14 | Two: 24½ | One: 18 |
| 12 | 21 | 18 | Rolled brass. | 21 | S.C. 30½, W.C. 31¾ |
| Wrought iron, in trunk. | Phosphor bronze. | Rolled phos. bronze. | 1¾ | Phosphor bronze. | Iron, brass cased. |
| 1¾ | 3 | 2½ | Centrifugal. | 3 | 3 |
| H. D. A. Vide air pump. | H. D. A. Vide air pump. | Centrifugal. | ... 10 | H. D. A. Vide air pump. | H. D. A. Vide air pump. |
| ... 14 | ... 22 | ... 8 | .. | ... 22 | ... 18 |
| 18 | 26 | ... | 8 | 26 | 30½ |
| 12 | 24 | 6 | 2–8 | 24 | ... |
| ... | ... | 2–0 | 14 | ... | 16 |
| 11 | 14 | 10 | Modified Griffiths. | 14 | Hirsch. |
| Modified Griffiths. | Hirsch. | Modified Griffiths. | Manganese bronze. | Hirsch. | Cast iron. |
| Manganese bronze. | Cast iron. | Manganese bronze. | 7–9 | Cast iron. | 17–0 |
| 10–6 | 14–3 | 9–9 | 9–4½ | 15–6 | 24–4½ |
| 13–2½ | 24–1 | 12–3 | Three. | 24–7 | Four. |
| Three. | Four. | Three. | 16 | Four. | 95.66 |
| 26.52 | 72.14 | 23.55 | 12.45 | 77.93 | 69.84 |
| 21.45 | 48.56 | 17.75 | | 54 64 | |
| 29½ 3–3 | 40½ 3–7 | 28 2–4 | 23 23 | 40½ 3–3 | 46 4–6½ |
| 4 | 7 | 3 | 3½ | 7½ | 9 |
| 2–6½ | 2–5 | 10 | 16 | 28 | 2–8 |
| 22½ | 2–5 | 22 | 18 | 2–8½ | 3–2½ |
| 1 to 1 257 | 1 to 1.689 | 1 to 1.28 | 1 to 1.21 | 1 to 1.58 | 1 to 1.42 |
| 1 to 2.542 | 1 to 2.232 | 1 to 3.83 | 1 to 2.03 | 1 to 2.095 | 1 to 2.766 |
| Four S.E. Low. | 2 D.E., 2 S.E., Scotch. | Two S.E. Low. | Four S.E. Low. | 5 D.E., 2 S.E. cylindrical | Eight S.E. Scotch. |
| ... 17–9 | 18–3 9–6 | ... 18–4 | ... 19–8 | 24–6 11–10 | ... 9–9 |
| ... 9–9 | 11–0 11–0 | ... 8–8 | ... 8–11 | 9–0 9–0 | ... 11–8 |
| ... 1⅛ | ¾ ¾ | ... ⅜ | ... ⅜ | ⅜ ⅜ | ... ⅞ |
| Steel. | Steel. | Steel. | Steel. | Steel. | Steel. |
| 12 Corrugated. | 12 Addison ring. | 4 Corrugated. | 12 Corrugated. | 12 External. | 16 Addison ring. |
| ... Three. | Four: Two. | ... Two. | ... Three. | One: Two. | ... Two. |
| 37 | 40 | 40 | 34 | ... | 48 |
| 5–9 | 6–9 | 7–0 | 5–9 | 7–0 | 7–0 |
| 220 | 270 | 93.2 | 195 | 672 | 400 |
| Wrought iron. | Drawn brass. | Wrought iron. | Wrought iron. | Wrought iron. | Drawn brass. |
| 2½ | 2⅜ | 2⅜ | 2½ | 3½ | 2⅜ |
| 6–11 | 6–1⅜ | 7–10 | 8–0 | 8–10 | 6–9 |
| 1,480 | 1,200 | 352 | 1,672 | 2,304 | 1,776 |
| 272 | 108 | 44 | 264 | ... | ... |
| No. 11 B.W.G. | No. 11 B.W.G. | No. 11 B.W.G. | No. 12 B.W.G. | No. 10 B.W.G. | No. 11 B.W.G. |
| No. 6 B.W.G. | No. 6 B.W.G. | 4⅞ | No. 8 B.W.G. | ... | ... |
| 3⅜ 3½ | 3⅜ 3⅜ | 1⅛ 3 11/16 | 3 2½ | 4½ 4½ | 4 3½ |
| 37.68 | 44.80 | 15.4 | 32.64 | 114 | 60.32 |
| 3–9 | 2–6 | 9 inches. | 4–6* | 36 D E. 33½ S.E. | 23 |
| 1/16 | | 1/16 | 1/16 | ½ F.C. ½ B.C. | 1/16 B.C. |
| ¼ | ½ F.C. ½ B.C. | ¼ | ¼ | ¼ | ¼ |
| 3/16 | ½ D.E. ½ S.E. | 3/16 | 3/16 | 3/16 | 3/16 |
| ½ | 1/16 | ½ | ½ | ½ | ½ |
| | ¼ | | | | |
| | ½ | | | | |
| Steel: 15 | Steel: 21 | Steel: 7 | Steel: 10 | Steel: 8 | Steel: 16 |
| 2½ | 1½ | 2½ | 2½ | 2½ | 2 and 2½ |
| 8,209.6 | 6,550.8 | 2,796 | 8,981.2 | 19,884.96 | 10,146.4 |
| 612 8 | 523.2 10 | 176 6 | 640 6 | 1,196.2 10 | 1,109.4 8 |
| 1,311 | 1,261.8 | 594 | 1,348 | 2,365.4 | 1,892.7 |
| 5.96 | 4.67 | 6.37 | 6 90 | 3 94 | 4.73 |
| 37.3 | 24.26 | 30 | 46.06 | 33.14 | 25.3 |
| 5 8 | 6.02 | 6.05 | 6.0 | 5.26 | 6.62 |
| 5.7 | 6.54 | 5.86 | 8.27 | 5.97 | 7.07 |
| 2.95 | 3.85 | 2.48 | 4.36 | 4.48 | 4.36 |
| ... 41.66 | 54.0 27.3 | ... 24.64 | ... 35.50 | 44.8* 21.5* | ... 32.97 |
| 2.68 tons. | 3.10 tons. | 2.365 tons. | 1.890 tons. | 2.70 tons. | 3.23 tons. |

of expansion, and the calculations show that in every case the jacketed engine is more wasteful than the unjacketed one. This is sufficiently evident, *a priori*, from the consideration that the latter receives all its steam at a maximum temperature, expands it adiabatically to a certain terminal temperature, and then exhausts it, while the former receives a part of its heat at intermediate temperatures, expands the fluid non-adiabatically, and finally rejects it at the terminal temperature with a lower mean range of expansion. In other words, the jacketed engine departs furthest from the principles of economical operation first enunciated by Carnot: All heat should be received at a maximum temperature; expansion should be perfectly adiabatic, and should continue to the minimum temperature and pressure, and all should be rejected as nearly as possible at that minimum. Thus, "theoretically," the unjacketed engine is more efficient than the jacketed engine.

"Practically," however, the reverse is usually, though probably not always, the case, and the use of the jacket is often found to be productive of a real and sometimes of large economy. It is thus obvious that the advantages of the employment of a jacket come of those conditions which distinguish so markedly the real from the ideal case in steam-engine economy.

Rankine's formulæ for a non-conducting and for a jacketed cylinder were used in working out the ideal cases:

*Non-conducting cylinder.*—Energy per cubic foot of steam admitted:

$$UD_1 = JD_1\left\{\tau_1 - \tau_2\left(1 + \text{hyp log}\,\frac{\tau_1}{\tau_2}\right)\right\} + \frac{\tau_1 - \tau_2}{\tau_1}L_1 + r(p_2 - p_3).$$

Heat expanded per cubic foot of steam admitted:

$$H_1D_1 = JD_1(\tau_1 - \tau_4) + L_1.$$

$$\text{Eff. of steam} = \frac{U}{H_1}.$$

*Cylinder jacketed.*—Energy exerted:

$$U' = a\,\text{hyp log}\,\frac{\tau_1}{\tau_2} - b(\tau_1 - \tau_2) + V_3(p_2 - p_3).$$

Whole expenditure of heat per pound steam:

$$h = J(\tau_2 - \tau_4) + a\left(1 + \text{hyp log}\,\frac{\tau_1}{\tau_2}\right) - b\tau_1.$$

$$\text{Eff. of steam} = \frac{U}{h}.$$

An initial pressure of 115 pounds was assumed in every case and a back pressure of 4 pounds, varying ratios of expansion being employed. The results are given in the following table:

| Cut-off. | Ratio exp. | Eff. without jacket. | Eff. with jacket. |
|---|---|---|---|
| 0.05 | 20.00 | 0.2073 | 0.1930 |
| .10 | 10.00 | .1934 | .1808 |
| .15 | 6.66 | .1795 | .1665 |
| .25 | 4.00 | .1566 | .1442 |
| .35 | 2.85 | .1358 | .1302 |
| .45 | 2.22 | .1237 | .1209 |
| .55 | 1.82 | .1119 | .1087 |
| .75 | 1.33 | .0898 | .0812 |
| 1.00 | 1.00 | .0707 | .0707 |

An examination of the table will show clearly the wasteful influence of the steam jacket as an element considered by itself. Within the usual range of practice, from about five or six to fifteen or twenty expansions, under the assumed conditions of initial pressure and cut-off, it is seen that the loss by its application is fairly constant at something above one per cent. in these cases, rapidly falling to zero as the ratio of expansion falls from the lower figures to unity. The consumption of steam in pounds per horse-power per hour may be computed very approximately by dividing 2.5 by the computed efficiencies. The cases assumed are for condensing engines; and the evaporation always taken at nine pounds per pound of fuel, the fuel expenditure may be gauged by dividing the weight of steam computed by nine. This gives, for example, about 12.06 and 12.95 pounds of steam required for the unjacketed and for the jacketed engine respectively at a ratio of twenty, and about 1.33 and 1.44 pounds of fuel. For a ratio of expansion of four the figures become about 16 and 17.3 respectively for the steam and 1.75 and 1.85 pounds for the fuel. At full stroke the figures become 35 pounds of steam and feed water and 4 pounds of fuel per horse-power per hour for both engines.

We conclude, therefore, that in the ideal case the steam jacket reduces efficiency, necessarily and without exception, and that for ordinary variations of the ratio of expansion for the pressure here assumed this waste is nearly a constant fraction. To put it mathematically, the gain is negative and nearly constant at about one per cent. efficiency.

(2) REAL EFFICIENCIES.—In 1886 a "Research Committee" was appointed by the British Institution of Mechanical Engineers to investigate the subject of the steam jacket. A very unusually complete set of data pertaining to trials made with a view to determining the efficiency produced by the application of the jacket was secured, and from these the following figures were collated and results deduced:

(1) The case of a single-cylinder non-condensing Corliss engine, 21.65 × 43.31 inches, the body only jacketed. The jackets were supplied by a small pipe from the main steam pipe and were automatically drained.

Following are the data as obtained, the experiments being carried on at variable boiler pressures and ratios of expansion:

SET I.

| Without jacket. | | | | With jacket. | | | |
|---|---|---|---|---|---|---|---|
| Boiler pressure above atmosphere. | Expansions. | Piston speed. | Feed water per I.H.P. per hour. | Boiler pressure above atmosphere. | Expansions. | Piston speed. | Feed water per I.H.P. per hour. |
| 110 | 6.2 | 445 | 28.38 | 110 | 7.1 | 439 | 22.80 |
| 110 | 6.2 | 445 | 28.38 | 110 | 6.2 | 447.5 | 22.13 |
| 110 | 5.0 | 443 | 26.82 | 110 | 5.3 | 447.5 | 22.35 |
| 110 | 4.4 | 459 | 27.58 | 110 | 4.4 | 453 | 23.65 |
| SET II. | | | | | | | |
| 78.2 | 5.4 | 447 | 27.58 | 78.2 | 5.1 | 441 | 23.65 |
| 78.2 | 4.8 | 440 | 26.71 | 78.2 | 5.1 | 441 | 23.65 |
| 78.2 | 3.7 | 433 | 24.58 | 78.2 | 3.8 | 445 | 22.58 |
| 78.2 | 2.9 | 437 | 24.18 | 78.2 | 3.1 | 437 | 21.95 |
| SET III. | | | | | | | |
| 49.8 | 3.7 | 443 | 27.27 | 49.8 | 3.9 | 437 | 25.30 |
| 49.8 | 2.5 | 441 | 27.20 | 49.8 | 2.8 | 437 | 25.21 |
| 49.8 | 1.7 | 440 | 30.17 | 49.8 | 1.7 | 440 | 28.74 |
| 49.8 | 1.0 | 437 | 46.82 | 49.8 | 1.0 | 441 | 46.26 |

The second case is that of a single-cylinder condensing engine

(Corliss), cylinder dimensions same as before, body only jacketed; experiments carried on at the same place, in the same year and in the same manner.

SET IV.

| Without jacket. | | | | With jacket. | | | |
|---|---|---|---|---|---|---|---|
| Boiler pressure above atmosphere. | Expansions. | Piston speed. | Feed water per I.H.P. per hour. | Boiler pressure above atmosphere. | Expansions. | Piston speed. | Feed water per I.H.P. per hour. |
| 110 | 13.5 | 433 | 23.24 | 110 | 11.3 | 424 | 17.05 |
| 110 | 10.7 | 423 | 22.24 | 110 | 10.0 | 444 | 16.65 |
| 110 | 8.2 | 429 | 21.41 | 110 | 10.0 | 432 | 16.50 |
| 110 | 6.4 | 416 | 21.99 | 110 | 6.4 | 419 | 17.59 |
| SET V. | | | | | | | |
| 88.9 | 9.3 | 432 | 21.19 | 88.9 | 12.0 | 430 | 17.66 |
| 88.9 | 8.7 | 428 | 21.05 | 88.9 | 11.3 | 430 | 17.32 |
| 88.9 | 7.3 | 432 | 20.79 | 88.9 | 6.8 | 429 | 16.87 |
| 88.9 | 5.9 | 419 | 19.89 | 88.9 | 5.9 | 419 | 17.50 |
| 88.9 | 4.8 | 427 | 20.43 | 88.9 | 5.9 | 433 | 17.50 |
| SET VI. | | | | | | | |
| 64 | 10.8 | 421 | 20.41 | 64 | 10.7 | 432 | 18.48 |
| 64 | 8.2 | 429 | 19.09 | 64 | 8.2 | 430 | 17.59 |
| 64 | 5.4 | 426 | 18.06 | 64 | 5.4 | 424 | 17.34 |
| 64 | 3.9 | 431 | 18.37 | 64 | 4.4 | 427 | 17.66 |
| 64 | 3.6 | 427 | 18.84 | 64 | 3.6 | 426 | 18.55 |
| SET VII. | | | | | | | |
| 49.8 | 10.7 | 438 | 20.65 | 49.8 | 12.0 | 435 | 19.29 |
| 49.8 | 7.9 | 424 | 19.36 | 49.8 | 7.6 | 416 | 18.48 |
| 49.8 | 7.9 | 424 | 19.36 | 49.8 | 7.6 | 416 | 18.48 |
| 49.8 | 5.6 | 436 | 18.77 | 49.8 | 5.6 | 432 | 18.17 |
| 49.8 | 4.2 | 424 | 19.00 | 49.8 | 4.0 | 434 | 18.91 |
| 49.8 | 3.2 | 424 | 19.78 | 49.8 | 3.2 | 432 | 19.67 |

The third case applies to a horizontal compound condensing tandem engine, the body of the cylinders only being jacketed. The whole steam supply to the cylinders passed through the jackets, which were drained by trap, and when not in use the jackets were open to the air.

The trials were carried on at a constant boiler pressure of forty-two pounds above the atmosphere, and a piston speed of 196 feet per minute.

SET VIII.—*Without Steam in either Jacket.*

| | | |
|---|---|---|
| Expansions, . . . . | 8.38 | 7.54 |
| Thermal units rejected per I.H.P. per min., | 525 | 533 |

| | | |
|---|---|---|
| Expansions, . . . . | 5.8 | 4.45 |
| Thermal units rejected per I.H.P. per min., | 619 | 574 |

SET IX.—*With Steam in H.P. Jacket alone.*

| | | | |
|---|---|---|---|
| Expansions, . . | 10.78 | 7.73 | 7.01 |
| Thermal units rejected, | 420 | 420 | 418 |

SET X.—*With Steam in L.P. Jacket alone.*

| | | | |
|---|---|---|---|
| Expansions, . . | 15.8 | 9.74 | 9.14 |
| Thermal units rejected, . | 427 | 380 | 388 |
| Expansions, . . . . | | 8.38 | 7.54 |
| Thermal units rejected, . . | | 383 | 385 |
| Expansions, . . . . | | 7.18 | 5.38 |
| Thermal units rejected, . . | | 407 | 427 |

SET XI.—*With Steam in both Jackets.*

| | | | |
|---|---|---|---|
| Expansions, . . | 13.2 | 13.1 | 8.38 |
| Thermal units rejected per I.H.P. per min., . | 347 | 348 | 375 |
| Expansions, . . | 9.14 | 7.54 | 6.7 |
| Thermal units rejected, | 367 | 372 | 385 |
| Expansions, . . . . | | 5.8 | 4.64 |
| Thermal units rejected, . . | | 370 | 410 |

The preceding results were plotted and a smooth curve obtained for each set, giving feed water used for different ratios of expansion both without and with the jacket. From the curves so constructed the feed water used at the same ratios of expansion with and without jacket was obtained, and the difference expressed in per cent. of the feed water used at that expansion without the jacket. These percentages were plotted, and gave efficiency curves for the jackets that take such a form as to indicate for each pressure used in the different cases the existence of a certain point of maximum efficiency corresponding to a certain ratio of expansion.

The feed-water curves having been plotted, the following values were obtained from them and used to plot the efficiency curves:

| | Ratio of expansion. | Efficiency of jacket, per cent. |
|---|---|---|
| *A* (Set I), | 6.2 | 21.33 |
| | 6 | 20.61 |
| | 5 | 15.79 |
| *B* (Set II), | 5.5 | 13.6 |
| | 5 | 12 |
| | 4 | 9.63 |
| | 3 | 9.38 |
| *C* (Set III), | 4 | 7.64 |
| | 3 | 7 |
| | 2 | 3.93 |
| *D* (Set IV), | 12 | 23.43 |
| | 11 | 23.82 |
| | 10 | 23.78 |
| | 9 | 23.2 |
| | 8 | 21.5 |
| | 7 | 20.32 |
| *E* (Set V), | 10 | 21.95 |
| | 9 | 22.14 |
| | 8 | 19.23 |
| | 7 | 16.66 |
| | 6 | 12.34 |
| *F* (Set VI), | 10 | 9.25 |
| | 9 | 8.62 |
| | 8 | 7.38 |
| | 7 | 5.94 |
| | 6 | 4.67 |
| | 5 | 3.6 |
| | 4 | 1.64 |
| *G* (Set VII), | 10 | 7.45 |
| | 9 | 6.28 |
| | 8 | 5.13 |
| | 7 | 3.84 |
| | 6 | 3.08 |
| | 5 | 2.03 |
| *H* (Sets VIII and IX), | 9 | 19.23 |
| | 8 | 20.45 |
| | 7 | 22.55 |
| *I* (Sets VIII and X), | 9 | 26.34 |
| | 8 | 26.34 |
| | 7 | 26.24 |
| | 6 | 26.5 |
| *K* (Sets VIII and XI), | 9 | 29 |
| | 8 | 29.35 |
| | 7 | 30 |
| | 6 | 31.62 |

Curves plotted from *A*, *B* and *C* show an increased efficiency with increase in ratio of expansion. Curves plotted from *D*, *E*, *F* and *G* also show that the efficiency of the jacket on that type of engine (single cylinder condensing) increases with increase in ratio of expansion, and in the case of *D* and *E* increases up to a certain value of the ratio of expansion and then decreases.

In the first case, that of the simple non-condensing Corliss engine, the heads unjacketed, taking the first example, the use of the jacket reduced the cylinder wastes from about twenty-five per cent. of the ideal consumption of steam and feed water to about half that proportion, for ratios of expansion approximating six; from one-third to about one-tenth for a ratio of five, and apparently from twenty to ten per cent. at 4.4. The same general effect is observed throughout, with some discrepancies, which may be due either to varying action of the jacket or to slight errors of observation or to both combined, the latter being the probable fact.

In the first case, also, it will be observed that the jacket gives best results, with 110 pounds of steam, when the ratio of expansion approximates six. When the steam pressure falls to approximately eighty pounds, the best work of the jacket occurs at a ratio not far from 4.75, while at the pressure of fifty pounds the value of the jacket increases through the whole range of the experiments, and not only so but the curve assumes a rectilinear form indicative of probable improvement indefinitely in the direction of increasing expansion. The highest efficiencies, however, either with or without the jacket, are found, in this case at the lowest ratios adopted, and indicate a maximum value at about 3.25. The ratios of expansion for maximum efficiency of fluid, in the other cases, are seen to be, for 110 pounds about five, and for eighty pounds about 3.5.

Similarly, studying the performance of the condensing engine, we find the best work done, whether jacketed or not, at about a ratio of expansion of ten (steam pressure 110 pounds), but that the jacketed engine reduces the internal wastes from fifty per cent. at highest ratios, and from one-fourth at the lowest ratios, in the case of the unjacketed engine, to five per cent.,

and, in some cases, probably to within the magnitude of the errors of observation. At a pressure of ninety pounds the best ratio seems to be for this engine, under the given conditions of operation about 6.5 when unjacketed and 8.5 jacketed; while the lower pressures still further reduce both the efficiencies and the savings effected by the jacket. The work of the jacket as an economizer of heat is done at a high pressure, at a ratio of expansion of twelve or more. In all cases it seems to be the fact, with these engines at least, that the jacket is useful beyond the ratios of maximum efficiency of fluid.

The compound engine is operated at altogether too low a pressure to bring out the best effect of compounding, but it exhibits the same general effects which have been noted in the cases of working of simple engines. The effect of the jacket is less pronounced than in the simple engine, and the efficiencies of fluid vary less with variation of the ratios of expansion. It gives its best results at ratios of expansion ranging from 7.5 to 10.5, the variations of the value being very much more observable in the last case, in which both jackets were in use, than in either of the others, and least in the case in which the H.P. jacket only was employed.

This discovery of a maximum efficiency of jacket may throw some light upon the causes of the conflicting and sometimes apparently irreconcilable results of trials of engines with or without jackets and with jackets variously constructed. The discovery may also prove of value to the designer, as aiding him in procuring the best proportions and arrangement of his engine.

# DISCUSSION ON VARIOUS PAPERS.

**Red-Hot Furnace Crown Experiments.**—CHIEF ENGINEER F. G. MCKEAN, U. S. NAVY.—The technical journals a few months ago detailed certain experiments in England, from which it was inferred that, as far as they had been carried, turning on the feed would be the best thing to do in the majority of cases when water is low in the boiler.

As the writer has seen no objection raised to this conclusion, except some deprecatory observations by the distinguished experimenters themselves, he would suggest that, whatever may be the case with the *Lancashire* boilers fitted with dispersion pipes in what becomes the steam space as soon as the water-level begins to fall, until experiments have been made with marine or other boilers which necessarily introduce the feed at a much lower level, we had better not act upon this inference.

The belief among most old engineers has been, that to start the feed into a boiler in which the water has become low for a sufficient length of time, would almost inevitably produce explosion, and the labors of the committee above alluded to do not seem entirely to controvert the old belief, though the experimenters may be correct in the inference that such explosions were not always due to the red-hot crown sheets. The reasons for only partial agreement with the deductions are the following:

1. When the experimenters declare that their boiler did not explode, we have no difficulty in admitting the fact, because we are told at the same time that the pressure fell in every instance in which the safety valves were kept open. The committee neglected to remind us why the pressure fell, and therein consists a vast difference between their special arrangements and those which obtain in marine and other practice. Owing to the high position of the feed pipes, the experiments simply instituted a simultaneous struggle between two steam producing furnaces acting with comparative slowness in the lower half of a boiler, and a steam-annihilating jet condenser acting rapidily in the upper half. The condenser was sure to win unless it was absurdly small, and its winning was doubly assured in this case by the pump confessedly supplying water enough for two or four boilers of the same size.

On the other hand, some old engineers will continue to believe, for the present at least, that while in actual service, in the majority of cases when the water is low, the feed would be introduced, if at all, near the bottom of the boiler, or at least below the actual as well as normal water-level, and, being cold, would remain there, displacing the boiler water already on the point of becoming steam; then lifting the latter, which would have a temperature of 230° and upwards, instead of from 44° to 62°, over the heated surfaces, which nowadays would be of steel instead of iron, and doing it slowly and without deluging them, the hot water would generate steam with more than normal rapidity until the weakest part of the boiler would give way in consequence of the increasing pressure, and half a dozen separate events would follow like flashes of lightning—as that genius, Zerah Colburn, explained—and the boiler would burst. The two cases are in several points distinct; let us avoid drawing universal deductions from either.

The reason for using so large a pump was theoretically excellent, as was also the reason for avoiding, near a public thoroughfare, the explosion which it was the ostensible object of the tests to risk, but practically, the former vitiated the experiments by intensifying the action of the *quasi* jet condenser, while the latter, together with other reasons, involved working with raised safety valves in all the experiments after 1*a*, a state of affairs which, to some extent, vitiates the conclusion, as we will endeavor to show presently.

2. The report lays great and repeated stress upon the steam beginning to fall in all the experiments with raised safety valves immediately upon the feed being admitted, but it passes more lightly over the fact that in the only trial with closed valves the pressure rose 21 pounds in $\frac{3}{4}$-minute. If the initial pressure, instead of being kept down to 6 pounds in this experiment, had been the normal for starting the engines, and 21 pounds had been added, to all intents and purposes instantaneously, the consequences might have been different in the experimental case, and would be so in many practical ones. Perhaps the observers, apart from the reasons given, and quite unintentionally, showed a certain consideration for their boiler when they carried out all subsequent trials with both safety valves blowing "hard," "freely," "fiercely," etc.

And before leaving the matter of safety valves, it is suggested that while the continuous escape of steam may be considered equivalent to its continuous flow towards an engine in motion, there are many cases in marine, and especially naval practice,

in which short stoppages are made with spread fires, or the fires kept banked for days together, with steam perhaps near the "top notch;" and in shop practice fires are often kept banked six nights in the week with safety valves down, *i. e.*, under circumstances similar to, but (with a leaky boiler and higher steam) more dangerous than those detailed in experiment 1*a*. It may also be suggested that if further trials are made the outflow from the safety valves, if they must be raised, deserves to be narrowly watched. In experiment 1 it was intended that both valves should be shut, but by an oversight one was left open. It seems as if the escape of steam from that one should have indicated: (*a*) a partial instantaneous fall of pressure, due to the partial vacuum around the sixty-four sprays of injection water, and which the length and small bore of the steam-gauge connections possibly made that instrument too sluggish to record; (*b*) an instantaneous rise of pressure, which the same sluggish records modified into an increase of only 6 pounds in 1¼ minutes, and (*c*) the self-evident, gradual fall of pressure as the steam was partly condensed and partly blown off.

3. The report implies in one place that it is hardly possible that explosions have been caused by pumping cold water on red-hot crown sheets, because the latter would have been rent, whereas in most cases the shell has given way. But this is an assumption. If the smaller, thinner, perhaps more perfectly circular cylinder with fewer joints and parts, happens to be more elastic or stronger than the larger, thicker shell built up of many parts, the former may stand in spite of overheating, and the latter may rend; while, if the water assumes the spheroidal state, it may be abnormally heated without, at first, being in actual contact with the red-hot plates. It is the pressure which starts the damage in any case, searching out the weakest spot in the whole structure, whether it be some "original malconstruction of the boiler or the dilapidated condition into which it has been allowed to fall," so that if the pressure is due in part to some other cause, the overheating of the plates may become of secondary importance, as, indeed, the committee implies in its allusion to the small specific heat of iron as compared with water. For "crown sheets" the marine engineer would probably prefer to read "top of back connection," for with really red-hot crown sheets he would most likely have heard previously from the tubes.

This may be a suitable place for allusion to the "ungovernable" pressure and the "instantaneous disengagement of an immense volume of steam," for which the committee has scorn and disbelief, though their steam gauge indicated a decided tendency

in that direction. In Zerah Colburn's theory of violent explosions one step was the momentary relief of pressure from the surface of the boiler-water, due more particularly to a leak in the case he was considering; and he argued that when this occurred the surface boiler-water, having a higher temperature than was due to the pressure, flashed into steam. Experiment 1*a* seems to corroborate this view, if we concede a partial vacuum due to the jets, followed by the instant conversion of the layer of abnormally heated surface-water into steam of some unknown pressure of which the gauge had time only to note a 21-pound rise, the further ascent being overpowered partly by the pump and partly by that very pressure again raising the boiling point above what was necessary for producing more steam. In the case of a rent in the boiler—and in practice that might have ensued from the typical 21-pound rise—all the boiler-water should theoretically have flashed into "ungovernable" steam; in the experiment it seems apparent why it did not. Critics who will not admit that a partial vacuum could have existed in any part of the boiler at any instant during the experiments 1 and 1*a*, causing the disengagement of a large volume of steam (because that should imply a greater supposable vacuum in the experiments with raised safety valves, due to the double action of an annihilation and a release of steam), must explain: (*a*) how a number of small streams of cold water can be injected into a large body of steam without condensing at least a part; (*b*) how steam can be condensed without forming an appreciable vacuum, and (*c*) if a partial and momentary vacuum *was* formed, why the unnoted entry of air at the same time that steam was escaping through the two safety valves in all the after experiments might not have assisted the newly forming steam to destroy that vacuum so quickly that the gauge was not affected at all in those trials, while it was measurably affected as to the rise of pressure in the first two.

We can infer that the crown sheets were decidedly hot in all the experiments, though perhaps not positively red-hot in more than one-half of them, and this brings up another point. In trials 1 and 1*a* there was low water "and no mistake," for it was blown down to 16 inches below the crowns; while in the timed experiments the water was evaporated off more slowly, so that its lowest level was about 2¾ inches below the crowns; it is a question if, in all of these later trials, a strip of metal on each side of the apex may not have carried off so much of the heat from the crown to the water below by conduction as to have lessened the effect of the red-heat, and in some cases annulled

it altogether. In the timed experiments the jet-action of the feed was limited, for it ceased as soon as the boiler water reached the level of the spring openings, and the red-heat, when it existed, was confined to a narrow strip; but the committee found that with unquestionably low water accompanied by greatly increased steam room, and with safety valves closed, there was a sudden accession of pressure when the feed was turned on. Now, whether that accession is attributable to a phase of Colburn's theory or to the exposure of a greater extent of furnace sheet, the rise of steam is certain, and throws a serious doubt upon the conclusion that in the majority of cases, with low water, turning on the feed seems to be the best thing to do, for it would appear to depend upon how "low" the water is, and we knew that before.

In another part of the report, however, it is acknowledged that perhaps one explosion in six does occur from pumping water upon red-hot crown sheets; also that nearly all explosions in confined spaces, (which would include those on board a ship) are fatal; and still it is suggested that turning on the feed would possibly be the best thing to do with low water! The information that there are six chances to one in our favor is a novel application of the factor of safety, and not yet sufficiently reassuring, we think, to make the practice universal.

The spheroidal condition of water has been alluded to by the experimenters, in one trial, for they say that the feed was continued until the plates had passed from red-heat to the temperature of water in the boiler. This is true, but in the account of that trial, No. 7, it is also stated that with two safety valves blowing "fiercely" and "freely," and with injection enough to "feed" two boilers, the pressure at one stage of this experiment only yielded at the rate of one pound per minute, insomuch that the valves were raised further by hand. Limiting our consideration to the question of pressure alone, suppose the safety valves seated, and only one ordinary feed, might not, nay, would not, the pressure have gone up? Then suppose the instantaneous and possibly "ungovernable" addition of some unknown pressure analogous to the 21 pounds noted above, might not the boiler have "gone up" too?

The published conclusions from the experiments can hardly fail to be dangerous, for the average water-tender who hears of them, not laying sufficient stress upon the declared doubts and cautions of the committee, will probably draw the inference that low water is a comparatively trifling matter; his induced carelessness will therefore be likely to bring it about, and when he

finds himself confronted with it, he will probably turn on the feed with a light heart, regardless of time and circumstances, and, in the absence of sufficient injection-roses, pumping-power and safety valves, disaster may be expected to follow in the future as in the past; for the implication in the report that because certain selected explosions specified were not caused by low water, therefore low water causes next to none, is a *non sequitur*, further based upon the possible fallacy that the hottest part of the boiler is necessarily, always and for that reason, the weakest part, and that crown sheets are the sole source of risk when the water has become low and feed is pumped into a boiler.

Finally, it is suggested that while the profession at large owes most cordial thanks to the committee for the time and trouble which they have taken, and the expense incurred, and should hope that the matter will not be allowed to rest under the slightest uncertainty, the experiments alluded to were not made under circumstances of universal application; that the sweeping inference drawn from them is misleading and dangerous, and that until the question is decided more fully, the marine engineer at least had better not "reinvigorate" with cold water the possibly hot steel plates of a boiler in service, even with the odds of six to one in his favor, or he may bring about an explosion without having the shelter afforded by a barricade.

**Trial Trips, &c.**—ASSISTANT ENGINEER W. D. WEAVER, U. S. NAVY.—The tests of indicators for the trials of the new cruisers demonstrated the fact that what has been considered an instrument of precision is very far indeed from being such, and this applies almost equally to the three standard types used, any difference being due rather to workmanship than to principle.

Notwithstanding that the indicators supplied were presumably of the best quality, some having been specially ordered for testing work, the errors they introduced in the indicator cards would probably average 5 per cent., and in some cases went as high as 10 per cent., and this refers only to the error due to the spring itself, those due to inertia and the drum spring and cord not having been considered.

The principal fault is one inherent in the present type of indicator, that due to the fact that equal increments of pressure do not produce equal increments of compression of the spring, and *vice versa*. It was found that if a spring was correct at the lower part of its reading it departed widely at the upper part, and if correct at the middle the errors gradually increased towards each end.

The tests developed a number of peculiarities in the action of indicators, and it is hoped that an account of these with a discussion of different methods of correcting indicator cards will appear in a future number of the JOURNAL.

In regard to the lubrication of indicators, it is doubtful if there is any necessity for it, as the condensation always present in the barrel should be sufficient for this purpose. The lubricating oil usually supplied with indicators is very detrimental at high pressure, as it is frequently carbonized by the heat. Several indicators were returned with the pistons stuck through this cause, and in one case it was with difficulty that the pistons of two indicators used on the trial of a launch engine were driven out with a mallet.

An electrical device for the simultaneous taking of indicator cards similar to that used by Mr. A. M. Mattice, and described on page 161, Vol. III, of the JOURNAL, was to have been used at the New York Navy Yard in taking cards for the purpose of comparing different methods of correcting indicator diagrams, and there should be no difficulty in applying it to any indicator.

In order to be able to reduce the effects of the inertia of the moving parts of indicators a search for a light alloy of aluminum sufficiently hard was instituted, but with no success. Different makers were written to, and an extensive correspondence otherwise carried on; several samples claiming to possess the desired hardness were tested, and the chemist of a noted aluminum company made various alloys, but in no instance was the hardness nearly equal to that of the softest brass. An addition of from 5 to 7 per cent. of some substance produced an alloy somewhat harder than pure aluminum, but the alloy became very brittle if the percentage of aluminum was less than 90.

As an instance of the "fad" in regard to steam passages, a contractor required of the manufacturers that the steam passage in the nozzles of the indicators used on a certain trial should be of the same diameter as the piston, notwithstanding that the displacement of the piston per stroke of engine was less than one-fifth of a cubic inch. Many "fads" were also developed by contractors' experts in testing. For example, it was insisted that the mercury column should be corrected for latitude. As the legal weight at any place is the weight of unit mass at that place, any other rule would require that a pound weight, for instance, for each place differing in latitude should be of a different mass; besides, there is no place accepted by convention to which all weights should be referred. Of course, for comparisons in most delicate investigations, weights are reduced to a common lati-

tude, but even were there a datum latitude, the contractor should be paid on the basis of pressures referred to the approximate local weight of the unit mass, not what that weight would be at the equator or at Paris.

In another case it was insisted that the temperature of the mercury column should be directly taken, as, owing to its different specific heat, it could never attain within four or five degrees of the temperature of the surrounding air, and only an experimental demonstration settled this point.

An accident at the trial of a boiler at the New York Navy Yard illustrates the danger of closed fire-rooms: a tube burst in the sectional tube boiler then on trial, and although a window sash was carried away by a fireman jumping through it to escape from the closed fire-room, the remaining fireman was dangerously scalded before he could reach the door, and Mr. Cowles, the designer of the boiler, who gallantly went to the rescue, was severely scalded in getting the man through the door. Had the same accident occurred in the fire-room of the *Cushing*, for example, every one there present would have been almost undoubtedly instantly killed.

Notwitstanding the knowledge of this fact, it was found that the firemen in the trial of the *Cushing*, after the first two watches, went about their duties as if they were in an ordinary fire-room, but, as on the trial of the *Baltimore*, an appearance of danger was a signal for a panic. In both cases there was a time when the water left the glass, and in both cases the men would have left the fire-room if they had not been restrained. Familiarity with modern steam generators will in time quiet apprehension, but in time of danger the presence of an officer, in whose professional knowledge the men will also have confidence, will be almost a necessity.

**Preservation of Marine Boilers.**—PASSED ASSISTANT ENGINEER G. W. BAIRD, U. S. NAVY.—I feel quite sure Mr. Leonard has not correctly estimated the relative lifetime of boilers in the Navy and in the mercantile marine. It will be found in the Report of the Chief of the Bureau of Steam Engineering for the year 1875, that the average lifetime of our naval boilers on board vessels having jet condensers was thirteen years and ten months, while it was but nine years in ships with surface condensers. Those averages were made by myself for the Engineer-in-Chief, and I extended my inquiries to some of the merchant vessels of New York, and found their boilers lasted but little longer than our own. The comparison cannot be properly made, for the reason that a merchant ship uses her

boilers continuously, while our vessels are put out of commission at the end of a cruise, sometimes remaining a year out of use. I could take only the averages for the times the vessels were actually in commission.

It is not generally known that distilled water is to some extent a corrosive fluid; it readily attacks the zinc on galvanized iron; copper is much more soluble in distilled than in hydrant or spring water. I have attributed the corrosion in the upper part of steam drums largely to the steam condensed there when the boiler was cooling.

About eleven years ago, while in charge of the machinery on board the ironclads at Washington Yard, I conceived the idea that the use of some hygroscopic substance inside the boilers would arrest the corrosion which had set in. I provided blank joints on the outboard pipes, when the valves leaked at all, to insure against any water entering. I provided tin vessels which would pass through the manholes and handholes which were to contain quicklime. My requisition for quicklime, however, was disapproved by a revising officer, as "quicklime is forbidden by the regulations to be brought on board vessels of the Navy." After a short period I obtained, on requisition, a quantity of the *oxide of calcium*, to which no objection appeared. After it had been placed inside the boilers of the *Montauk* for about ten days, I removed the manhole plates and found the lime well slaked. I replenished the pans, again closing the boiler, and on subsequent examinations found that the boilers were perfectly dry inside, and that the lime showed no appreciable hydration. Subsequent to this I learned the same thing had been done in England with similar results.

I note Mr. Leonard's reference to pitting in the bottoms of the drums. The original boilers of the *Vandalia* were provided with cylindrical drums, placed longitudinally in the spandrels above the boilers, and unless especially drained these drums would contain condensed water to the depth of about two inches. The drums "pitted" very badly. I collected a phial of the water from one of them and a chemist found copper in it. I suppose the copper came from the brass in the linings of the air pump, from the condenser tubes, valve seats, etc.

In regard to Mr. Leonard's remarks about carbonate of lime scale, I would say that it sometimes forms into a hard mass. In this building (the State, War and Navy Department building,) we have boilers aggregating about 500 horses-power, all of which are fed with Potomac water. In summer time, when there is not much rain, and when the river is fed by the springs,

which are laden with carbonate of lime, we find our boilers scale up quite fast; the scale appears exactly like that we get in our boilers at sea and is quite hard. The analyses show the scale to be nearly all carbonate with but little sulphate of lime.

I have heard surprise expressed at the absence of dampers from our ships built during and since the war. We used dampers up to the time of the civil war; at that time the Navy was increased suddenly by the addition of purchased steamships and new war vessels as rapidly as they could be built. We did not have a large mercantile marine to draw from, and a number of engineers of but limited experience had to be employed. The Engineer-in-Chief therefore determined to eliminate every part that could be dispensed with, make the designs as simple as possible and with but few parts. Independent cut-offs, which had been thought indispensible, were discarded; a single pump with the salt water on one side of its piston and the fresh water and vapor from the condenser on the other, was the air and circulating pump, and it was worked from the main engine. The damper was dispensed with at the same time.

Since the introduction of the mineral cylinder oils, I think the deterioration of boilers has diminished. A black deposit from this oil is frequently formed in the boiler near the water level, and so far as I have observed, has been a means of preserving the boiler at that part, though I can readily see how it would be a dangerous deposit on the heating surfaces. However, much of it can be arrested by the employment of a good filter. My belief is, that by a judicious use of proper cylinder oils, the employment of a good filter, and replenishing the waste water by an evaporator, the period of existence of our marine boilers may be doubled.

The importance of Mr. Leonard's paper cannot be overestimated. The high powers demanded, and the scant weight allotted for our machinery, has led to our shaving down the scantlings in our boilers to a minimum. The boilers are covered by heavy armored decks, and are placed in cells made of the many water-tight bulkheads. The cost of taking a boiler out of such a ship will probably be greater than the shop cost of the boiler. The durability of the boiler has therefore become a serious question, and the necessity for heavier shells—shells which may last as long as the hull—will, I hope, in the near future be considered.

**Engine-Room Signals.**—CHIEF ENGINEER JOHN LOWE, U.S. NAVY.—I very much regret my enforced absence from the last meeting of the Society, for the subject up for discussion, Engine-Room Signals, is one in which I am intensely interested.

That system known as Navy Bells I regard with a species of horror, and I never miss a proper opportunity, official or unofficial, to hit that system with all my strength.

I regard it as a device of the Evil One, because looking upon it on dress parade its appearance is good; but in supreme moments, like its master, it serves no other purpose than to betray good men and good ships into trouble. Personally I have never made a cruise without having been brought into trouble by it. The trouble with the system is this, that it introduces the personal element too largely. A man of iron nerve can, perhaps, under most circumstances ring the navy bells; but all men are not of iron nerve, and that is the point: mistakes are exceedingly liable to occur.

Let me illustrate: Conceive a vessel going at, say, 16 knots—that is to say, four bells—and let a sudden danger arise, making it necessary to go astern as quickly and as powerfully as possible. Concede the officer of the deck to be of iron nerve. He will placidly pull, first one bell, then he will placidly proceed to mentally measure a proper interval, and afterward pull two bells. Next, maintaining his equanimity, he mentally measures another interval, and then placidly pulls three bells; now then another interval, and then another three bells, after which the requirements are satisfied. If the man at the throttle valve is alike unruffled, why, then, the signals have all been answered, and meanwhile the ship has or has not overtaken her fate, as the case may be.

These men of iron nerve are to be found, and they are very comfortable men to deal with; but men of other temperaments are likewise to be found, and the chances are that under the emergency I have pictured the signals, either given or understood, will have been hopelessly mixed. Too much has been expected from human nature, and too much has been given it to do under a sudden strain.

Now, if you please, that system may have been fitted with a recording device which will show what the mistakes were. That serves a very useful purpose for a court-martial, but it does not prevent the mistake nor the damage. In like manner the signals may have been transmitted by means of air, or by means of water, or by a wire. These are only means or intermediates, and do not affect the main question. Now, what is the remedy? The first step to be taken is this: Cease living in a fool's paradise, and recognize the fact and keep it always present in the mind, that in the navy moments of intense terror may be expected, in which even the best men are for an instant unnerved.

Having taken this in, proceed to give them an instrument to signal by, with which even in such moments the tendency will be to go right, and not wrong.

Such an instrument is to be found in the annunciator, so called, when well fitted up. It may be operated by means of a wire, by compressed air, or by means of water. It may also have a recording device. All these are not to the purpose. The point gained is, that the operation of signaling is made simple instead of complex.

To be properly fitted the following points should be rigidly adhered to: When *ahead* is signaled the transmitting handle points ahead and in no other direction; when *astern* is signaled the handle points astern and in no other direction; when *stop* is signaled the handle must point vertically upward and in no other direction.

The dial is to contain the following signals and no others: AHEAD—*full speed, half speed, slow, stop; slow, half speed, full speed*—ASTERN. All other signals can conveniently come down the voice pipe.

Under these circumstances the captain, even though in the crow's nest, can, by the mere extension of his hand, indicate to the transmitter his wishes, and be understood without a word being uttered, and by night as well as by day. The position of the handle may be felt if it cannot be seen. In the engine room the same rules are to be rigidly adhered to, and in addition the annunciator must be placed within easy reach of the man at the throttle. In supreme moments the weight of his responsibilities has a tendency to keep this man's nerves steady; and he is consequently the best man to observe, repeat and obey the signals.

In at least one vessel with which I am acquainted these rules have been departed from in this particular. The annunciator is placed upon a transverse bulkhead, and the man at the throttle must leave his post and return to read, answer and obey the signals. In ordinary times no trouble is to be feared, but let a moment arise requiring more than ordinary haste, can it not easily be seen that two men, intent upon their own purposes, meeting at this point may effectually block the way of the man at the throttle?

In order, then, that the signaling apparatus shall be properly fitted, the man who has to do with its installation at the building of the ship must have certain requirements to follow. In order to meet them he must have the right of way. After the signaling apparatus is in place, steam gauges, clocks, counters and the like may be fitted, but the bell man must be considered first.

There is one point in Chadburn's system that is a good one. The go-ahead gong is of one tone and the go-astern gong is of another, and thus by the sound the man at the throttle may operate first, and afterwards, to reassure himself, he may look at the dial and answer the signal. Thus the annunciator system may take advantage of the good points of the ordinary bell system as well. When the man at the throttle has obeyed the signal and has answered it the officer of the deck has the information he desires. Any other device or instrument is therefore wholly unnecessary and in many ways undesirable.

CHIEF ENGINEER F. G. MCKEAN, U. S. NAVY.—The Society is to be congratulated upon the selection of such an important subject for discussion. The prompt transmission of engine-room signals may save the ship.

The following notes have reference only to the signals made by gong, either when that is the sole means used, or when the other methods may be out of order.

The writer believes that he was the cause of an order being promulgated that the repetition of a signal means the intensifying of it, *i. e.*, while running under one bell, the repetition of one bell would emphasize slowness, and therefore mean *slower*, and not, as has been claimed, produce non-action at the throttle upon the plea that "We are already going slow." Similarly, the repetition of three bells would emphasize backing, and would mean *back stronger;* and four bells while in full course would signify the very best that can be made, regardless of working off steam, but understood to be carried out only for a few moments in avoiding collision, closing up to maintain position, crossing a mud-bar, etc. It is important that the number of signals be as small as possible to avoid confusion, and our naval system is preferable to that in the merchant service, as it does away with the "jingle-bell" with its extra wires and attachments.

If movements are needed in the opposite sense from those specified above, there is scope for them already; *e. g.*, in squadron tactics, if it be required to go faster, but yet under one bell, there is no need of a messenger, even in the absence of a speaking-tube, for four bells alternating with one will either produce the required speed after a few attempts, or the mean of the two speeds will keep the ship in position; but the people on deck should bear in mind that in close order in line ahead, if our predecessor stops or backs suddenly, it is not enough that our ship should do the same, but the signal should be at once announced to others by the steam whistle, because an instant of inattention

on the part of the following ship might cause our being rammed. Signals by whistle would probably be available in all cases, even that of a squadron in a fog, where the relative positions are maintained, for example, by the first ship blowing one blast; the second blowing two; the third, three, and so on, because mistake could be avoided by the blasts of position being moderately long, while those of manœuvre should be short and rapid.

But to return to the Engineer Department proper. The writer suggests that on every ship it should be a recognized part of the routine duties to examine every bell-crank, chafing-spot, and length of wire at frequent, stated intervals; and also that this should be one of the understood preparations for getting under way: a leisurely turn of the pliers upon a slack length, or the timely removal of a pair of socks hung in a dark corner to dry, may avert disaster. It is also suggested that in spots where the wire might give way from local causes a spare piece be secured firmly by one end and wound loosely around the suspected portion, so that if the original wire parts a new one is half attached and ready to bridge the gap.

In conclusion: if nervous individuals can only be convinced that a steady pull of a pound or two is generally sufficient, whereas a tug with both hands, as if one's dearest friend was being rescued from a fiery furnace, may disarrange the system, it may be hoped that "Engine Room Signals" by bell will still be suitable, at least as being supplementary to the later systems which appeal more directly to the eye.

Passed Assistant Engineer W. F. Worthington, U. S. Navy.—I am glad the Society is going into this question, and offer my experience for what it may be worth. The *Atlanta* is provided with a mechanical telegraph, marked Chas. Corey & Son, Pat. No. 60,502, and the ordinary engine-room gong. The telegraph is good as far as it goes. It is used exclusively when working the engines, the speaking tube being used to regulate the number of revolutions. It has been used a great deal in the last two years and five months, and during that time the index has never failed to indicate correctly, except on one occasion when one handle was secured and excessive strain was put on the other, thus breaking the chain. The signal bell attached often fails to strike, but that does not occasion much trouble, because the dial is close to the man at the reversing wheel. I would suggest the following improvements:

(1) That a series of rods jointed by bevel wheels or by Hooke's universal joints take the place of the chain, the joints all to be easily accessible for inspection. The chain cannot be

inspected, as it is in a closed tube, therefore defects cannot be discovered until there is a failure.

(2) That a bell be used that can be relied on to strike every time the index moves.

(3) That provision be made for signaling single revolutions, having recourse, if necessary, to two or more pointers connected like the hands of a clock, and indicating units, tens and hundreds respectively.

(4) That a pencil be connected with the index by suitable levers and made to travel parallel to the axis of a drum carrying paper. When the pencil touches the paper and the drum is made to revolve by clock work it is obvious that every movement of the index would leave a permanent record, giving the exact instant when that movement took place.

This apparatus should be supplemented by a revolving index in the chart house, permanently connected with the main engines by a series of jointed rods, as described before for the mechanical telegraph. The deck officer would soon learn to judge approximately the speed by observation. He could determine the exact number of revolutions at any time by applying to the end of the shaft (of the revolving index) a Heath's Patent Self-Timing Self-Registering Speed Indicator. I only recommend this crude apparatus for immediate use, until some one can invent a tachometer which will work and give the exact number of revolutions.

The Baird apparatus, to show the direction of motion of the engine as fitted in this ship, although very useful, is defective in three respects:

*a.* It does not show the number of revolutions.

*b.* It does not start simultaneously with the engine.

*c.* It is driven by a belt, which slips occasionally, as all belts in an engine-room will, from steam and grease, in addition to other causes.

The old fashioned engine-room gong fitted on this ship is rarely used except in port, to give warning when the running boats are about to leave.

The greatest defect of the gong is, that if from inattention or preoccupation with other work, the men in the engine room fail to catch the signal, there is no way by which they can be sure what the signal was. Also they sometimes think they counted the bells right and are mistaken. This latter occurs when at critical times a great many bell signals are made in quick succession.

As the gong is fitted in this ship, there is no way by which

the deck officer can tell if the bell struck at all, if it struck correctly, or if the signal has been heard by the man in the engine room.

I should strongly object to the use of electricity for signaling to the engine room, or for showing in the chart house the speed of the engine. The electric lights and bells on board frequently fail to operate without giving any previous warning of their defective condition.

The electric apparatus for showing when the double bottoms contain water has never worked, although many of the compartments have at times had large quantities of water in them. The electric apparatus for signaling when a coal bunker is on fire has never worked but once, and on that occasion there was neither fire nor even anything combustible in the bunker. But the fundamental difficulty with all such electrical apparatus is, that it is impossible to tell by inspection whether it is certain to work or not.

On the other hand, with the jointed rods and bevel gear used on this ship to work the valves of the steam-steering engine from the chart house, and traversing a distance of about 300 feet, no failure has ever occurred within my experience (two years and five months), and the friction is so slight that a man in the chart house can steer the ship with *one finger*.

PAST ASSISTANT ENGINEER G. E. BURD, U. S. NAVY.—The Council's invitation to discuss the engine-room signal apparatus leads me to express some views picked up during my cruise with this "Squadron of Evolution." Probably more use of engine-room signals has been made during this cruise than ever before in the same length of time. I have been shipmates with three of the "annunciators," as now furnished, and they have given much trouble and are generally out of adjustment, or partially so. It seems almost impossible to keep the system of wires and chains of the proper length by reason of stretching, change of temperature or some other cause, and the chains frequently slip off the wheels when turning corners. They have also necessitated stationing a man, from the already insufficient number of men on watch, to work the handle in the engine room during the execution of manœuvres, which generally last all day while cruising. The bell is not loud enough to be heard if the machinist should happen to be in another part of the engine room. The only gain on the old gong is that the person making the signal on deck has an answer that the signal is received. With a good shaft indicator this is hardly necessary, and, as the old gong has greatly the advantage in simplicity, I think most sea-going officers would prefer it.

As to the shaft indicators, I have been with an electric indicator worked from a small shaft driven through gear wheels by the main shaft. It was always getting out of order, as a delicate electrical apparatus might be expected to do when exposed to the heat, ashes, steam and other impedimenta found in an engine room. Theoretically it was excellent, as it gave both direction of motion and number of revolutions of shaft. It is easy to design some apparatus that will work wonders if sufficiently complicated, but if it be placed in an engine room it will have found only a short stopping place on its way to the scrap heap.

I have been with five tachometers, and they have been allowed to remain in place on account of their being somewhat ornamental, but none of them have ever been of the slightest use. The same is true of these patent recording gauges and other mechanical curiosities that are placed in engine rooms. Such things are usually highly finished and handsome to look at, and, as they are ingenious, they take the fancy at first; but in my experience they are worthless. If a result can be accomplished by simple means let us have it, but with the large amount of necessary machinery which a few engineers have to look out for on a modern ship, complicated additions are not apt to receive the attention they demand. It is found very difficult to keep even the incandescent lights going in the engineer's department, owing to the unfavorable conditions to which they are exposed there, and I believe that electric signals, counters, etc., would fare much worse.

We have on this ship a Baird's shaft indicator, which works well and never requires any looking after, but it shows only the direction and not the revolutions.

The speaking tubes between the engine room and pilot house have been the main vehicle by which orders have been transmitted.

When in squadron, making ordinary runs, it is intended to make a constant speed and for each ship to keep in a certain fixed position with respect to the other ships. To this end the flagship sets the speed, and the revolutions of her engine are taken and averaged every ten minutes, and the speed thus kept fairly regular. The other ships try to keep a fixed distance from their leading ship, either by estimating the distance or occasionally taking a mast-head angle. If we fall slightly behind, the order is given through the tube to increase the speed of engines, say from 40 to 43 revolutions. The machinist opens the throttle slightly and begins to count the revolutions. (Our tachometer pointer is vibrating through about 20 to 30 revolutions and so is

useless.) Before he has time to count this half minute the officer of the deck, not noticing an immediate increase in the speed, gives the order to increase to 46 and soon after to 50 revolutions. The ship now begins to gather increased way, and rapidly overtakes the leader, when the slowing-down orders are given till the engines are slowed much below the mean speed, and the ship falls behind again, when the same process is repeated. This is the ordinary way of running, and necessitates the constant attention of the machinist on watch, who is thus kept from his legitimate duties. He always fails to give satisfaction, and gets frequent messages from the deck that he is not paying attention to his work, because he cannot make the speed instantly correspond to a slight change in the throttle. The fact that the native language of the men at the ends of the tube is often different sometimes adds to the general dissatisfaction of all concerned.

The result of this system is that the time of the machinist is entirely taken up in answering calls from the deck and the engines do not receive from him the attention which is their due, and which ought to be his most important duty; the services of another man are lost to the engineer's force by his having to be stationed at the annunciator; the men of the engine-room force are harrassed and confused by the unceasing stream of orders from the deck and their inability to satisfy the officer of the deck; and, lastly, the officer of the deck is convinced that the ship's being out of position is caused by the engineer's force not attending properly to their duties.

The whole system, then, is most unsatisfactory to all concerned. But I believe the whole trouble may be overcome by a most simple and inexpensive method, which I will give at the risk of being thought prolix and irrelevant.

I would have a small shaft fitted with mitre gears to turn corners, and worked by a wheel in the pilot house similar to the wheel for steam steering. A clutch in the engine room would be thrown in or out of gear with an efficient balanced throttle valve. A double poppet valve like the *Yorktown's* throttle would work well, I think, but its stem should screw in and out like a stop valve instead of being moved by a lever, as in the *Yorktown's* valve. It would thus be capable of fine adjustment, and a man who can steer well would soon become able to regulate the speed without sudden changes when the rest of the squadron is in plain sight to him. The stop valve should be in a handy position and should be set, under the direction of the engineer on watch, so that if the throttle be wide open it will give a speed to the engines equal to that for which the order

was given and for which boiler power was put on. In case of priming or other accident the machinist could either close the stop valve or throw out the clutch and close the throttle, informing the officer of the deck, through the tube, of his reasons for so doing. There is no question about the practicability of such an arrangement, as far as the mechanical part is concerned, for a similar set of shafting works the valve of the steering engine of this ship in a perfectly satisfactory manner and requires no attention or repairs, and the steering engine is twice as far from the pilot house as the engine room. This system I consider less liable to derangement than a wire-rope connection, and I have had experience with both. This would leave the machinist of the watch free to attend to his legitimate duties of looking after the engines, and the responsibility for being out of position would be fixed where it belongs.

I would also have a lever with a notched arc, similar to the reversing gear of a locomotive cab, in the pilot house, which, being properly connected to a lever in the engine room and by that to the reversing engine, would enable the officer of the deck to stop or back if called to do so suddenly in an impending collision or otherwise. I have planned a reversing gear which might have this connection and at the same time allow the machinist in the engine room to stop, back, or link up, without disconnecting the deck attachment or otherwise losing time, no matter in what position the deck lever may be set. Whether this connection should be used ordinarily or only in case of emergency would depend upon the results obtained; but I believe it would become usual for the engines to be worked from the bridge after they are turned over and reported ready. A bell signal should be fitted for use in case of accident to the gear.

In answer to the question as to what system I would prefer, I would answer that I would have no system of signals except for auxiliary purposes, and have the engines worked from the deck. In the absence of such a system, I would prefer a simple gong with the old system of signals and a speaking tube. Then when the officer of the deck strikes two, followed by three bells, all the people in the ship will jump up to see what is the matter.

I have come to the belief that a simple thing, of which not so much is expected, is much better than one of great promise and a liability to break down.

PASSED ASSISTANT ENGINEER R. S. GRIFFIN, U. S. Navy.—I think Mr. Norton's suggestions are very good ones, and that their adoption would obviate much of the confusion that now exists on account of the dials not being similarly engraved in each

engine room. On the *Philadelphia*, motion ahead is indicated by the pointer moving against the hands of a watch for the starboard engine, and with the hands for the port engine; and as each dial occupies the same position relatively to the reversing engine, behind and to the left of the operator, confusion is almost inevitable if the same person attempts to work both engines, as is sometimes necessary. Motion with the hands of a watch for the ahead motion would remedy this, and adopting differently colored backgrounds for the ahead and backing divisions of the dial would serve as a check. As to the order *stand by* for backing, I don't see the necessity for it: if the ship is getting under way somebody will aways be at the reversing engine, and if she is under way the order to *stand by* would indicate all that is necessary, either that the engines would be stoppęd or backed. I hope we haven't any reversing engines which require so much nursing to back as would necessitate a special preparatory signal. And if any change of cut-offs is considered necessary, the one order *stand by* should be sufficient for the necessary adjustment.

PASSED ASSISTANT ENGINEER H. MAIN, U. S. NAVY.—I entirely agree with Assistant Engineer Norton as to the expediency of all his suggested changes in marking the dial faces of the annunciators. And as the mechanically connected annunciator has proved fairly successful and is readily adjusted, there seems no present need of resorting to the uncertainties of the electrical instrument. I wish to add a few words as to the location of the engine-room dial, the importance of which was emphasized to me while on duty on the U. S. S. *Vesuvius*. Between the starboard and port engines of the *Vesuvius* is a narrow fore-and-aft passageway, at the end of which were placed the two annunciator dials and gongs. The gongs were duplicates with the same tone, and the dials having brass faces and being in a dark place at some distance from the reversing levers could not clearly be seen by the man working the engines. Someone, consequently, had to stand by and give his whole attention to answering and then facing about and passing the signals to the machinist working the engines. This is bad enough for a single engine, as in case of the *Boston* or *Atlanta*, but when there are two engines and two annunciators placed close alongside, the liability to mistake in transmission is greatly increased, as we soon found out.

As subsequently arranged, the annunciator dial for each engine was placed directly over the reversing lever for that engine with the handle for the answering signal in easy reach of the machinist working the engine, the brass faces of the dials were frosted, making the letters distinct, and the two gongs were made

of very different tones. The facility and quickness in working the engines were remarkably improved and the possibility of mistake reduced to a minimum, as the man who worked the engine himself first moved the dial handle to answer the signal. This requirement, I think, should always be provided for in locating the annunciator in the engine room. It may be objected that the moving of the dial handle takes a perceptible time where quickness is most essential on the part of the man working the reversing lever or wheel, but it will be found invariably that this time is less than the time else required to hear and act on the signal as called out. Incidentally there is also, of course, an economy of engine-room attendants.

## NOTES.

### ELASTICITY OF BACK TUBE SHEETS.

In a letter from the Orlando Bros., published in the London Engineer of May 1, they call attention to a method of construction adopted by them in fitting the back tube sheets, to overcome the difficulties with leaky tubes when using forced draft. They believe with Mr. Yarrow that lack of elasticity is at the root of the evil, and to insure this they secure the tube sheet in place through the intervention of a U-shaped copper ring. This construction was followed on two locomotive boilers. One was worked at 140 pounds pressure, with 5 or 6 inches air pressure, without any trouble whatever; the other, working at 180 pounds steam pressure, was steamed continuously for eight or ten days with air pressures of 2 or 3 inches, with continuous variations in the fire, and no leakage occurred. They consider that the reason why the boilers of locomotives give no trouble while the same type fails on torpedo boats, is because the greater length of tubes of the former gives the requisite elasticity.

They believe that the trouble had with common combustion chambers is due to the same cause. In some six furnace common combustion chamber boilers they are building there are for each

square foot of grate surface about 4 square feet of direct heating surface, including tops of furnaces, combustion chamber and tube plates, while in locomotive boilers this ratio stands about 1 : 3. It is consequently difficult to explain why a common combustion chamber that has not to absorb an amount of heat equal to that absorbed by a locomotive fire-box, should leak from overheating of its tube plates, especially steaming at a lower air pressure. Probably when the direct heating surface is small in proportion to the grate surface, as in the case of the *Barracouta*—something less than three times—a certain amount of overheating may take place and facilitate the sliding of the tubes in the tube plates.

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PURVE'S RIBBED FLUES.

The superiority of the Purve over the ordinary or the spirally corrugated flues is owing greatly to the processes pursued in its manufacture, which are much less distressing to the material than those required in the manufacture of its rivals. The ribs are formed on the plate in the rolls. The plate is planed on the edges perpendicular to the ribs and bent into shape. For forming the weld of the butt joint short pieces of the best iron are inserted, the two plain cylindrical portions on either side of a rib being welded before the rib is welded, the iron welding strip being here fitted with special care. Heat for welding is supplied by portable furnaces, which heat the metal both from the inside and the outside. After the welding is complete, which is accomplished by a power hammer, the anvil and the hammer being shaped to conform with the ribs, the flue is carefully annealed in a vertical furnace and is finally trued and straightened by being passed through hydraulic dies fitted to the ribs.

The height of the ribs, measuring from the outside of the plain portion, is uniformly $1\frac{3}{8}$ inches, and the distance between them is also uniform, 9 inches. The minimum thickness of the plain portion is $\frac{7}{16}$ inch. This is increased at the ends by $\frac{1}{8}$ inch to allow for thinning incidental to flanging.

The following comparative table, giving the pressures allowed by Lloyd's Register and the British Board of Trade, shows the extent to which the Purve flue is given the preference:

| Kind of Flue. | Lloyd's Formulæ. | Board of Trade Formulæ. |
|---|---|---|
| *a*. Ordinary corrugated. | $p = \frac{1000(T-2)}{D}$. | $p = \frac{12500t}{D}$. |
| *b*. Spirally corrugated... | $p = \frac{912(T-2)}{D}$. | $p = \frac{11100t}{D}$. |
| *c*. Purve's ribbed..... | $p = \frac{1160(T-2)}{D}$. | $p = \frac{14000t}{D}$. |

$p$ = maximum pressure per square inch.
$T$ = thickness of flue in $\frac{1}{16}$ of an inch.
$t$ = thickness of flue in inches.

$D$ for Lloyd's formulæ = greatest diameter of flue for cases *a* and *b*, and the outside diameter of the plain cylindrical portion in case *c*.

$D$ for the Board of Trade formulæ = the mean diameter for *a* and *b*, and the outside diameter of the plain portion for *c*.

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## LOFTY SMOKE PIPES.

[From the "London Engineer."]

It may be safely said that the builders of the new Cape steamer *Scot*, the Messrs. Denny Brothers, broke new ground when they decided to fit that vessel with funnels one hundred and twenty feet high. The loftiest chimneys previously used at sea were, we believe, those of the *Great Eastern*, the tops of which were one hundred feet above the grate bars. Hitherto seventy-five to eighty feet has been regarded as rather an extreme height. It is not very easy to find a reason why short chimneys should be used, now that masts and sails have become secondary considerations.

No doubt at one time it was deemed expedient to keep chimneys as short as possible, because for some reason best known to sailors it was held that a tall chimney was objectionable. But the sailor has very little to do with the modern high-speed steamship, and if lengthening a funnel gives a good draft instead

of a bad one, it seems to be foolish to permit nautical prejudices to stand in the way.

The only real objection that can be urged, we think, is that a long funnel is more likely to be blown overboard than a short one. But the loss of a funnel is one of the rarest accidents imaginable, and while wire rope is plentiful and cheap, there is no reason why a funnel should not be made as secure as the mainmast of a clipper ship spreading a cloud of canvas.

The objection which has been raised against the tall funnel, that the wind pressure against it created a measurable resistance to the progress of the ship, is avoided in great measure by making the funnel a flat oval in section, with the major axis fore-and-aft; and it must not be forgotten that by increasing the length of a chimney its diameter may be reduced.

While, then, the objections to the use of lofty chimneys appear to be more sentimental than rational, there can be no doubt concerning the benefits which plenty of chimney height confers. An early example of this was supplied by H.M.S. *Bellerophon*, which on her trial trip failed to realize the intended speed. Among other alterations made, the funnels were lengthened ten feet, with a direct and considerable improvement in the draft of the furnaces.

When the products of combustion escape at a low temperature a point is soon reached beyond which little is to be gained by adding to the height of the chimney, but at sea the chimney temperatures are unfortunately great, and within possible sea limits, we may go on adding to the height of a steamer's chimneys with the certainty that each addition will make the draft better.

Rankine says that "to insure the best possible draft through a given chimney the temperature of the hot gas in the chimney should be nearly, but not quite, sufficient to melt lead." This condition is pretty nearly fulfilled in steamships. There is good reason for believing, however, that this will hold good only of furnaces which deliver directly into the chimney, but that for steam boiler furnaces there is practically no limit to the temperature that will give the best draft.

As to the height which will give the greatest possible draft, that cannot be reached at sea, and it only remains to be considered what height is necessary to give a certain rate of combustion per square foot of grate per hour. Direct experiment has shown that to burn thirty pounds per square foot of grate per hour we must have a difference of pressure between the ash-pits and the uptake of about .75 inch of water. Under some circumstances and with some coal less will do. We may take it for granted then that if a chimney is long enough and large enough, and the gases within it are hot enough to give .75 inch of water pressure, every reasonable demand has been satisfied. Approximately, the rate of combustion, other things being equal, varies as the square roots of the heights of chimneys; but this must not be taken as a hard-and-fast rule. There appears to have been no difficulty in getting, at sea, in the *Scot* an air pressure equal to about $\frac{5}{8}$ inch of water with her chimneys of 120 feet in height, and it is possible that, were the chimneys a little larger in cross section, a still better draft could be had. With the long chimneys we are able to get many, if not all, of the benefits of assisted draft, while the fire rooms are open and cool. It is true that a certain advantage gained by heating the air, as in Mr. Howden's system, is forfeited.

The question then remains for consideration: Which is commercially the most advantageous, to run fans and heat the air required for combustion, or to trust entirely to length of chimney, to the exclusion of all complication in the shape of fans and engines? This is a question which cannot be answered off-hand, and deserves careful discussion and consideration. The long chimney is the simplest possible means of obtaining a good draft, and it will be surprising if the example thus set by the builders of the *Scot* is not followed, so providing sufficient data to enable the required comparison to be drawn. It will be worth while, however, to collect information as to the performances of furnaces with various heights of funnels at sea.

# CASUALTIES AND REPAIRS.

### FRACTURE OF A CRANK PIN ON THE GERMAN LLOYD STEAMER *LUBECK* ON HER TRIP FROM APIA TO SIDNEY.

On January 10, the intermediate engine suddenly developed a heavy thump. At the time the steam gauges recorded a pressure of 155 pounds in the boilers, and 54 and 11.5 pounds in the first and second receivers, respectively. As the thumping grew momentarily worse, the engine was stopped and examination showed that the crank pin of the I.P. engine had broken square in two near the forward crank-web. Four separate attempts were made to repair the damage, and in the end the steamer succeeded in making her port under reduced power about four days over time.

1. To establish direct communication between the H.P. and L.P. cylinders, the slide valve of the I.P. cylinder was removed, the I.P. piston blocked up at the upper end of the cylinder, and the connecting rod disconnected. The fracture of the pin severed the crank shaft into two pieces, and the gap was bridged by bolting pillow-block caps against the flat opposite sides of the I.P. webs, the bolts passing through between the webs. The caps were of rectangular cross section. The bolts used were the cap bolts of the I.P. crank shaft bearings. The caps had to be bolted up in an oblique position to allow the bolts to clear the webs. The pillow-block caps removed for making these repairs were replaced by wooden ones.

Ten hours sufficed to do the work, and the engine was started at 42 revolutions. After about five hours' running they had to be again stopped, as the connections became slack. During this time the pressure in the boiler was 114 pounds, in the receiver 57.

2. The holes in the middle of the caps for feeling the journals were enlarged to allow the bolt to enter, and the caps secured in a horizontal position. The time occupied was seven hours, and the engine was again started at 42 turns.

After six hours' running the bolt became slack and the engine was stopped and the nut screwed up. After this the engine ran

for sixteen hours making 30 turns. At the end of that time the caps had worked themselves out horizontally, so that there was danger of striking the engine foundation. The engine was stopped and the following additions made:

3. Four turns of $\frac{5}{8}$ inch chain were passed around each crank web and the caps. The space between the chain and the webs was filled with wooden wedges which served to tauten the chain. The engine was started but immediately stopped, as it was apparent that the caps still worked out sideways. The space between the bolt and the crank webs was now wedged up with pieces of iron and wood. This forced the fractured surfaces apart, and they were brought close up again by screwing up on the thrust collars. Of these latter three were set to bear on the ahead going faces of the rings and two on the backing faces.

4. The material used in (3) for filling the space between the bolt and crank webs proved unreliable. Two spare crosshead slippers were therefore trimmed off to a fairly good fit, the spaces remaining filled with hardwood wedges, and the whole wrapped with strong rope. The engine was now started at 38 turns. Several days' running proved the connection secure, the revolutions were gradually increased to 42, and the ship was enabled to reach Sidney without further mishap.

### ACCIDENT TO H. M. S. *SEAGULL*.

A most singular accident which occurred to the machinery of the torpedo-boat catcher *Seagull* during her full-speed trial on the 26th of June is described and illustrated in the issue of "Industries" for July 3d. On that day the engines, at the beginning of the trial, were gradually worked up to 200 revolutions per minute, and appeared to be working fairly well; but when indicator diagrams were taken it was found that the port low-pressure cylinder was doing practically no work at all. At first it was thought that perhaps the indicator gear was out of order, but a few experiments demonstrated that the difficulty lay deeper than this. After consultation it was decided that the most probable cause of the defect was that the balance-ring at the back of the slide valve was out of order. This valve is of the usual flat double-ported type, except that the exhaust passes direct from the center of the valve to the exhaust pipe.

It was thought that probably the vacuum ring had become stuck to the body of the valve through the softening of the india-rubber ring which encircles the joint between the ring and the valve. However this might be, it was clear that the port engine was working as a compound engine simply and not as a triple-expansion one.

Nothing serious enough to cause an injury to the mechanism seems to have been surmised, so the trial was proceeded with, the port engine being allowed to work at an easier rate than usual.

After the trial and return to port an examination was made, when it was discovered that the *main valve had been fractured into several pieces, and these pieces had been blown into the exhaust pipe and condenser*, nothing having been noticed by those in charge of the machinery. Two large pieces of the valve were found in the exhaust pipe, and several smaller ones on top of the condenser tube plate. The two larger pieces weighed some twenty pounds apiece.

## BOOK REVIEWS.

THE MECHANICAL ENGINEER'S REFERENCE BOOK. BY NELSON FOLEY AND THOMAS PRAY, JR.

We have carefully examined this work, and can pronounce it a most excellent reference book for the use of marine engineers. The number of reference and handbooks intended to convey information in a condensed form and make designing easy is legion, but there is no one which can be said to be all that is wanted. The work under consideration comes nearer the mark than any other that we know of, and its form is admirable, reducing to a minimum the time and labor required to find what is wanted. The graphic representation of calculated results is a valuable feature. "The Mechanical Engineer's Office Book—Boiler Construction," by Nelson Foley, embodied in the present work, has been in use in the drawing room of the Bureau of Steam Engineering, and its worn condition is the best recommendation that could be desired. The book is for sale by the D. Van Nostrand Co., 23 Murray St., New York, who are the sole agents for the United States.

# SHIPS.

## UNITED STATES.

*Cruiser No. 13.*—Bids for this vessel were opened June 1. The appropriation was limited to $2,750,000. The bidders were: The Bath Iron Works, at $2,690,000; Wm. Cramp & Sons, $2,745,000, and the Union Iron Works, $2,793,000. After much delay the contract has been finally awarded to Wm. Cramp & Sons.

*Cushing.*—The following data and results were obtained in the progressive trials of the *Cushing*, of July 20 and 21 ultimo, the runs having been made on the Potomac river under the supervision of a Board of Naval Engineers.

Complete data and discussion of results will be given in a subsequent number.

The letters *a*, *b*, *c*, *d*, heading the columns denote the 1st, 2d, 3d and 4th hours of the test, respectively. It is proper to remark, in view of the falling off in the speed developed, as compared with the contractor's trial, that these progressive trials were not intended to reach a maximum performance, but solely for the collection of data in a series of progressive tests within easily definable limits, besides which is the additional fact that the engineer force is considerably less than that neccessary for a maximum result.

These trials are the first in which the horse power has been obtained.

| | *a* | *b* | *c* | *d* |
|---|---|---|---|---|
| Steam pressure at boilers, per gauge, in pounds | 84. | 114. | 191.9 | 222.5 |
| Steam pressure H.P. steam chest, per gauge, pounds | 65. | 98.9 | 173.3 | 218.8 |
| Steam pressure 1st receiver, absolute | 34.5 | 51.3 | 91. | 110. |
| Steam pressure 2d receiver, absolute | 13.9 | 22. | 43.3 | 51.5 |
| Steam pressure 3d receiver, absolute | 7.4 | 10.1 | 17.1 | 20.8 |
| Vacuum in inches of mercury | 25.3 | 24.1 | 23. | 19.9 |
| Revolutions per minute, starboard engine | 205.9 | 240.1 | 298.8 | 328.3 |
| Revolutions per minute, port engine | 199.3 | 234.2 | 285.4 | ..... |
| Air pressure in inches of water, forward fire-room | ..... | ..... | 1.6 | 2.0 |
| Air pressure in inches of water, after fire-room | .4 | 1.7 | 1.2 | 2.1 |
| Grate surface in use, square feet, | 38.3 | 38.3 | 76.6 | 76.6 |
| Total I.H.P., both main engines | 261.30 | 449.52 | 926.54 | 1,272.60 |
| Speed of boat in knots per hour, by observation | 13.08 | 13.4 | 15.42 | 17.5 |

*Torpedo Boat No. 2.*—The following corrections should be made in the table published in No. 2, Vol. III., of the JOURNAL, giving the principal dimensions of the boat as designed by the Department: The displacement is 120 tons; the beam, 15 feet 6 inches; the stroke of the engines is 16 inches; the diameter of the L.P. cylinder, 30 inches.

## ENGLAND.

*Melampus.*—On the 12th of June the *Melampus*, cruiser of the *Latona* class of 3,400 tons displacement, made an eight hours' full-power trial of her machinery under natural draft.

The result was satisfactory so far as the engine and boiler performance was concerned, but at maximum speed the vibration of the hull was excessive, so great, indeed, that "fiddles" or "racks" were necessary to retain dishes on the table. This cruiser was built and engined by the Naval Construction and Armaments Company of Barrow, this company having also built the *Latona*. The maximum I.H.P. recorded was 8,080, with a mean for the

eight hours of 7,656, the average of the revolutions having been 135 per minute. Boiler pressure 147 pounds, vacuum 27.5 inches.

The propellers are three-bladed, with a diameter of 13 feet and a mean pitch of 17 feet.

On the 17th of June the *Melampus* completed her four hours' full-power trial under forced draft with the following results:

Boiler pressure, 146.7 pounds; vacuum, 26.8 and 26.1 inches; revolutions, 140 and 141 per minute; collective I.H.P., 9,093, being 93 I.H.P. in excess of contract. The average air pressure in fire room was 1.28 inch. The average speed, by log, was 18.35 knots per hour.

*Vulcan.*—The *Vulcan* is a ship constructed with the object of sending her wherever an English war ship may be, thoroughly equipped for making all sorts of repairs and fittings to the torpedo gear. She is to carry torpedo boats, and act as a floating workshop for a fleet.

The principal dimensions of the *Vulcan* are as follows:

| | |
|---|---|
| Length, | 350 feet. |
| Beam, | 58 feet. |
| Displacement, | 6,620 tons. |

She will carry six second-class torpedo boats, one vedette boat and two ordinary service steam launches.

The main engines, which are from the shops of Messrs. Humphreys & Tennant, are of the triple-expansion type, with cylinder diameters of 40, 59 and 88 inches, and a common stroke of 4.25 feet. There are two sets of engines driving twin-screws, and the aggregate I.H.P. to be developed is 12,000, on a boiler pressure 150 pounds, the engines to make 100 revolutions a minute. This result was to be obtained with forced draft of 2 inches air pressure. Under natural draft the contract called for 7,200 I.H.P., maintained for twelve consecutive hours.

The main condensers, of which there are two, are of composition, as are also the tubes.

The circulating pumps are driven by independent engines.

The total cooling surface in the main condensers is 13,500 square feet.

There is one auxiliary condenser having its own circulating pump and engine, into which all the auxiliary engines exhaust.

The ship is provided with five boilers, four double-ended, each containing six furnaces, and one auxiliary boiler, single-ended, with three furnaces. The double-ended boilers had each originally one common combustion chamber for the six furnaces, but it was found necessary to erect brick partitions, dividing the combustion chamber into six parts. The furnaces have also been shortened by bricking up the ends of the grate bars, reducing the effective grate surface from 600 square feet to 488 square feet.

The workshop on board contains five lathes, two shaping machines, two drilling machines, slotting, shearing and punching machines, a circular saw, a blacksmith shop, moulder's shop with hot-air furnace, and some twelve vises.

The workshop machinery is driven by its own engine, and it contains, in addition, a dynamo driven by one of Willans' engines.

On the 23d of April last a trial was made of four hours' duration, the mean I.H.P. developed being 8,166, and the machinery performing satisfactorily. The speed obtained was 18.5 knots.

During the progressive trials to which the *Vulcan* has been subjected during the last few months many defects in hull, machinery and boilers have been developed, and it seems that so far from being the success her first trials indicated, the ship is regarded as a failure, the hull being deficient in stiffness for the strains incident to her duties, the engines showing signs of weakness at any attempt at extending the original horse-power developments, and the boilers having been in a chronic state of leakiness and uncertainty at every attempt at forcing their performance.

## FRANCE.

*Le Tage.*—One of the most important recent additions to the naval force of France is the first-class protected cruiser *Le Tage*, built for the government by contract at the yard of "La Société Anonyme des Ateliers de la Loire."

Her principal dimensions are as follows:

| | |
|---|---|
| Length, | 389.75 feet. |
| Beam, | 53.5 feet. |
| Depth, | 35.92 feet. |
| Draft, mean, | 22.83 feet. |
| Displacement at this draft, | 7,045 tons. |

The hull is constructed of steel; the stern, stern-post, keel and protective deck are, however, of iron. The protective deck extends the full length of the ship, and is placed below the water line. A belt of cellulose extends from end to end of the ship on both sides.

The engines are of the three-cylinder triple-expansion horizontal type, actuating four-bladed twin screws of manganese bronze. The cylinders are of the following dimensions: H.P., 43 inches; I.P., 68 inches; L.P., 100 inches, with a common stroke of 47.25 inches. The low-pressure cylinders are forward, or nearest the boilers, and the crank shafts are arranged with couplings for disconnecting, enabling the engines to work as ordinary compound by putting the low pressure cylinder out of use. The H.P. and I.P. cylinders are steam jacketed. Joy's valve gear is employed.

Steam is supplied by twelve boilers arranged in three groups in water-tight compartments. These boilers are 14.25 feet in diameter, 10.79 feet in length, and each contains three of Fox's corrugated furnaces 3.58 feet in diameter and 7.58 feet long. The total area of grate surface is 930 square feet. With the exception of the 792 stay tubes, which are of iron, the boiler tubes, numbering 4,608, are of brass. These tubes are of 3 inches inside diameter, the iron stay tubes being but 2.625 inches diameter. There are three smoke-pipes 8.25 feet in diameter, one for each group of four boilers.

There are two surface condensers, one in each engine-room, the total cooling surface being 20,194 square feet, equal to 1.75 square feet per I.H.P. with forced draft, 2 square feet per I.H.P. with natural draft, and 8 square feet per I.H.P. when the engines are working as ordinary compound. Each condenser contains 5,633 tubes .7 inch in diameter and 9.66 feet long. All pumps are independent of the main engines. There are two air pumps

in each engine-room, single-acting, 28.75 inches in diameter by 19.72 inches stroke. A centrifugal circulating pump is located in each engine-room, having a disc of 48 inches diameter. There are two single-acting feed pumps in each engine-room having cylinders 6.75 inches in diameter with a stroke of 19 inches.

The main crank shafts are 15.75 inches in diameter, with a 7-inch axial hole.

The forced-draft engines are compound, the blowers delivering air into closed fire-rooms at a pressure of from 1 to 1.5 inches of water.

The valve gear is arranged to cut off in the H.P. cylinder from 25 to 74 per cent. of the stroke, in the I.P. cylinder from 25 to 70 per cent., and in the L.P. cylinder from 37 to 79 per cent.

With natural draft, and with all the cylinders in use, carrying boiler pressure of 150 pounds per square inch above the atmosphere, the I.H.P. developed was 8,950, revolutions 89 per minute. Under forced draft, with 97 revolutions per minute, an I.H.P. of 11,370 was developed.

The guaranteed speed is 19 knots per hour for six consecutive hours, the consumption of coal not to exceed 33 pounds per square foot of grate per hour. For each $\frac{1}{100}$ knot in excess of this speed the contractors receive $2,000, forfeiting a like amount for each $\frac{1}{100}$ knot the speed falls short.

The coal capacity of *Le Tage* is 900 tons.

The total weight of machinery in detail is given as follows: engines, 540 tons; shafting, 72 tons; propellers, 16 tons; boilers, with attachments, 489 tons; water in the boilers, 250 tons, a total of 1,367 tons.

The contract price was $1,684,320, divided as follows: hull, $871,200; armor, &c., $111,808; engines, $532,400; boilers and appurtenances, $207,345.

### GERMANY.

The first of four battleships now building for the German government was recently launched from the government works at Wilhelmshaven. The other three ships are building by con-

tractors, two at the Vulcan Co.'s works at Stettin, the other at the Germania Co.'s works at Kiel. These ships are not nearly as far advanced as the one building at the government's works. Battleship "*D*," the one just launched, was named the *Kurfuerst Friedrich Wilhelm*. She is built of steel throughout, of German manufacture, and everything about the ship, excepting only the anchor hoisting engine, is of German material. The stem, stern-post and shaft struts are of cast steel furnished by Krupp. The stem consists of three separate pieces, of which the middle one weighs 33,000 pounds. The ship has peculiar but graceful outlines, with sides having a deep tumble-home amidships and a flaring bow.

| | | |
|---|---|---|
| Displacement, | 10,000 | tons. |
| Length over all, | 380.5 | feet. |
| Beam, extreme, | 64 | feet. |
| Draught, | 24.27 | feet. |
| Depth of hold to upper deck, | 43.3 | feet. |
| Height above water, | 19 | feet. |

The ship is built on the longitudinal bracket system. The double bottom, the wing passage bulkheads, and numerous transverse bulkheads divide the ship into 120 water-tight compartments.

The armor consists of a continuous belt of compound armor 15.75 inches thick, on a teak backing. The width of the belt is nearly uniform throughout. It is worked up against the ram, which is immensely stiffened thereby. A protective deck 2.56 inches thick protects the vital parts.

The armament consists of six 11-inch guns in three turrets; six 4.13-inch Krupp rapid-fire guns, built on an entirely new system, in broadside and protected by light armor; eight 3.43-inch guns, disposed chiefly for firing ahead and astern, and two lighter R.F. guns in the military tops, besides a number of revolving cannon and a torpedo outfit.

The propelling machinery consists of two sets of triple-expansion engines in separate compartments; I.H.P., about 9,000; speed estimated at 15 knots. Steam is furnished by twelve

cylindrical boilers in three separate compartments steam; pressure, 175 pounds; grate surface, 753 square feet; heating surface, 23,400 square feet.

There is a balanced rudder, well protected. The ship carries two military masts and has two smoke-pipes.

### GERMAN NAVAL FORCE.

In 1889 a program was adopted for the building of a powerful modern navy, the ships to be completed by 1895.

The following table gives the number and character of the ships provided for, also the number that may be expected to be ready in 1895. The work has been much delayed by the failure of the Reichstag to vote the annual appropriation. Some of the ships already completed have failed to make the speed they were designed for. The 10,000-ton battleships were originally intended to have a speed of 18 knots, but this was subsequently changed to 15.

| | According to the program of 1889. | | According to trials made, the work in progress, and the money voted. | |
|---|---|---|---|---|
| | Number. | Speed. | | |
| 1. Heavy battleships, 10,000 tons | 4 | 18 | 4 | 15 |
| 2. Small battleships, 3,800 tons | 10 | 18 | 6 | 16 |
| 3. Protected cruisers of from 4,500 to 5,500 tons | 10 | 20 | 3 | 18 |
| 4. Small unprotected cruisers, 1,500 tons. | 7 | 16 | 6 | 16 |
| 5. Torpedo cruisers, 780 to 850 tons | 3 | 24 | 3 | 18 to 20 |
| 6. Torpedo dispatch boats, 350 to 450 tons | 10 | 20 to 26 | 10 | 20 to 26 |
| 7. Seagoing torpedo boats, 85 to 150 tons | 100 | 20 to 26 | 100 | 20 to 26 |

### DENMARK.

A crucial test was made to determine the water excluding properties of cellulose. The new cruiser *Hecla*, which has a water-line belt of cellulose three feet thick, was selected for the test. Another naval vessel, the *Absalom*, approached within

about 110 or 115 feet, and sent a shot from a 5-inch gun into a paralellogram drawn upon the *Hecla's* sides. The shot passed through the ship and out at the opposite side. Immediately after the shot was delivered the *Hecla* got up anchor and steamed about for 3 hours at the rate of 16 knots, the water reaching well above the holes. Yet at the termination of the trial only about 2 feet of water were found in the compartment affected.

## MERCHANT STEAMERS.

*City of Paris.*—This fine ship having been thoroughly overhauled ran her trials on May 1, and on May 13 began her regular trips.

The starboard engine, which suffered such complete collapse during the accident which occurred a year ago, was entirely rebuilt on the old plans and patterns. Such of the old parts as proved strong and sound were used again. Of the high-pressure engine there were found suitable for use the piston, piston rod, connecting rod, eccentrics, eccentric rods, valve motion, air-pump lever and air pump complete, sole plate and one of the supporting columns. Of the intermediate pressure engine, the piston rod, connecting rod, eccentrics and valve motion, with one column. The low-pressure engine is entirely new. The shafting is new from end to end. The shaft bearing in the after bracket was lengthened 9 inches, making it 7 feet, although its length was considered sufficient before. As an evidence that extraordinary circumstances caused the wearing away of the starboard bracket bearing of the *City of Paris*, it is worthy of note that on the *City of New York* being docked recently, it was found that neither the port nor the starboard bearing had worn in an appreciable degree after eleven voyages. A change was made in the method of fitting the lignum-vitæ. There is a strip of brass between the strips of lignum-vitæ, and the wood is placed with the grain end on against the shaft. The wood had

been previously fitted on the Admiralty system, which consists in inserting the wood in staves without interstices, the whole being prevented from turning, and the grain of the wood being parallel to the shaft.

To enable the level of the tube shaft to be accurately determined at all times, a simple device has been applied, already at work on the *City of New York*. It consists simply of a tube passing through the ship's counter directly over the shaft axis and fitting into a cup in the bracket. A measuring rod dropped into this tube and resting on the shaft indicates on a graduated scale any wearing down that may take place.

A controlling arrangement has been connected with the Brown steam and hydraulic starting engine, which stops the main engines when the revolutions exceed a certain maximum, 120 in the present case. In addition to the ordinary steam and hydraulic cylinders there is a supplementary steam cylinder, which has steam always on the top side of the piston. A weighted bell crank, which can be adjusted to any number of revolutions, is kept reciprocating by a small lever attached to the air-pump lever. When the limit is reached in the number of revolutions, the inertia of the weight causes the upper arm of the bell crank to engage in a detent on the valve stem of the small steam cylinder. This cylinder moves the fulcrum of the float lever controlling the steam and oil valves of the starting cylinders a sufficient distance to bring these valves into a position corresponding to the mid-position of the link. The latter arriving in this position, the differential gear again closes the steam and oil ports, leaving the link in mid-gear. On trial the controlling gear was set to stop the engines at 80 revolutions, and on the speed being intentionally accelerated the gear did its work quickly and effectively. The Messrs. Thomson fitted a similar arrangement years ago on the *Gallia*, but instead of being regulated for a certain number of revolutions it was actuated from a Dunlop governor, the action of which decided at what speed the engine was to be brought to a standstill.

The change in the steam-generating plant has been the substitution of Howden's system for the closed stokehole system pre-

viously in use. The furnace bars were shortened 9 inches, reducing the grate surface from 1,293 to 1,026 square feet.

Air is furnished by twelve Sturtevant blowers placed on the main deck. The fans are 6 feet in diameter, and are driven at 430 revolutions by double engines 7 and 7 × 5. A single piston valve furnishes steam to both cylinders. The cranks are opposite. The whole engine is completely enclosed, but all parts are accessible when the door is opened.

The nine double-ended boilers are arranged in three water-tight compartments in sets of three. The blowers draw their air through ten trunks, five on each side, the forward and after ones on each side supplying one blower, the others two each. Each fan discharges downward through a square trunk between the main and lower decks, from the bottom of which a rectangular air pipe is led downwards, and thence along the front of the uptake of the boilers immediately above the top of the boiler tubes. The fans on the starboard and port sides of the ship discharge respectively into the starboard and port ends of this pipe. There is a rectangular opening at the vertical center line of each boiler to allow the air to pass into the heater. This latter extends nearly across the whole front of the boiler, and consists of vertical iron tubes 3 inches in diameter and 34 inches long, the bottom tube plate of the heater being from 1 to 2 inches above the upper row of boiler tubes. From the heater the air passes down at each side of the outer smoke boxes to an air reservoir which encircles the upper half of the furnace front, occupying the whole space upward to the bottom of the smoke boxes, from which it is separated by an air-tight sheet iron casing. From this reservoir the air is admitted by valves to the closed ash-pits and to the space between the inner and outer furnace doors, from which it enters the furnaces through $\frac{3}{8}$ inch holes in air distributing boxes. The space between grate bars is $\frac{3}{8}$ inch. At the sides of the furnace are put plates fitting closely into the corrugations.

The temperature of the air in the reservoir is generally about 210°. On the trial on the measured mile in the Firth of Clyde, made on May 1, the following results were obtained: Average revolutions of fans 370 per minute; air pressure in reservoir $1\frac{1}{2}$

inches; above the fire and in the ash-pits, $\frac{3}{8}$ of an inch; steam pressure, 143 pounds; vacuum, starboard 27, port 26; revolutions of main engines, starboard 86, port 86.1; mean speed, five runs, 21.047 knots; mean speed, four runs, 20.968 knots; I.H.P., 18,003.28; draught forward, 21 feet 9 inches; aft, 22 feet 4 inches. Propeller, 19 feet 6 inches diameter, 28 feet mean pitch.

The results are below those obtained on the original trial, when the speed was about 21$\frac{3}{4}$ knots; but the ship drew 9$\frac{1}{2}$ inches less, and the displacement was 500 tons less; probably also, the lack of experience of the stokers with Howden's system had something to do with reducing the efficiency, and lastly, the ship's hull required cleaning. Mr. Howden is confident that the boilers will be able to furnish all the steam the engines can use. They have been previously run up to 90 revolutions. At any rate the opportunity will be afforded of testing the relative efficiency of the two systems of forced draft, as the engines are now the same as when the boilers were worked under the closed stokehole system.

*S. S. Illinois.*—This ship and her sister, the *Pennsylvania*, has been changed from a passenger to a cargo steamer. The old compound engines and the old boilers, which together occupied nearly one-third of the hold, were taken out, and a set of triple-expansion engines and a single high-pressure boiler were put in instead. The main boiler is designed to carry 160 pounds. Two auxiliary boilers placed over the main, and carrying 80 pounds, are used for cargo purposes, running the dynamo, auxiliary machinery and for making up the water supply. Howden's forced draft is used, a 60-inch Sturtevant blower supplying the air. A pressure of 2$\frac{1}{2}$ inches is carried in the conduit. Temperature of blast, about 200° F.; temperature of escape gases, about 600° F. The principal dimensions of propelling machinery are given below. The accompanying indicator cards and table give results obtained on the six hours' sea trial:

*Engine*, one inv. cylinder, 3-X; cylinders, $\frac{23'' \times 36'' \times 60''}{36''}$.

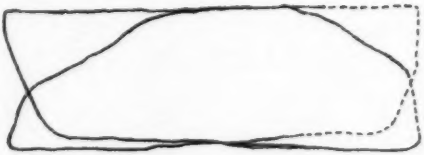
*Screw*, 4-bladed, true; 14 feet 7 inches diameter, 17 feet pitch, 53.28 square feet hel. area.

Indicator diagrams. taken of Main E

S/S ILLINOIS during trial trip on

H.P.C. Eng. No 105 Scale 100

Card No 3.

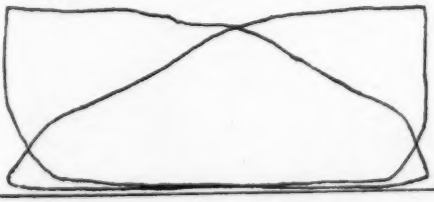


Atm. line

| Pressu |
| --- |
| " |
| " |
| Vacuu |
| Revol |

M.P.C. Eng No 105 Scale 50

rd No 3.

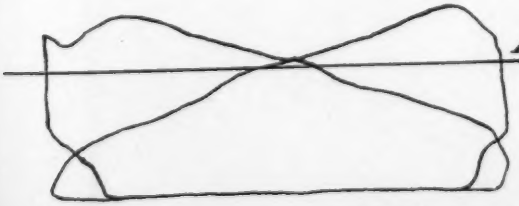


Atm. line.

| | |
| --- | --- |
| Diameter | |
| Stroke ... | |
| Mean Eff. Pressure | |
| Ind. H.P. ... | |

L.P.C. Eng. No 105 Scale 12

rd No 3.



Atm. line.

l Main Engines of

l trip on May 22nd 1891.

## Memorandum.

| | |
|---|---|
| Pressure in Boiler..... | 158,5 lbs. gage. |
| " " 1st Receiv. | 60. " " |
| " " 2nd | 4 " " |
| Vacuum..... | 25 ins. |
| Revolutions per min.... | 76.5 |

| | H.P.C. | M.P.C. | L.P.C. | |
|---|---|---|---|---|
| iameter | 23 | 36 | 60 | ins. |
| troke ... | 36 | 36 | 36 | ins. |
| ian Eff. Pressure | 67.2 | 38.15 | 8.37 | lbs. |
| d.H.P .. | 387.74 | 540.00 | 329.11 | Total 1256.85 IHP. |

*Boilers*, 160 pounds limit on main. One main boiler, 15 feet diameter, 11 feet 7½ inches long; 3 furnaces. Heating surface: furnace, 142.96 feet; tubes, 1,960.88 feet; fire box, 240.04 feet; total, 2,343.88 square feet. Grate surface, 57 feet.

Two auxiliary boilers, steam 80 pounds limit. 10 feet diameter, 8 feet 6 inches long; 2 furnaces each. Heating surface, each: furnace, 56.94 feet; tubes, 597.89 feet; fire box, 87.17 feet; total, 742 feet; total for both, 1,484 feet. Grate surface, each, 29.16 feet; for both, 58.32 feet.

One Sturtevant blower, 60 inches diameter of fan; 450 revolutions per minute.

*Maximum performance*, main engines: revolutions, 76.5 per minute; steam, 160 pounds per gauge; vacuum, 25 inches; I.H.P.: H.P., 387.74; I.P., 540; L.P., 329.11; total, 1,256.85.

Auxiliary boilers used only for dynamo and blowers.